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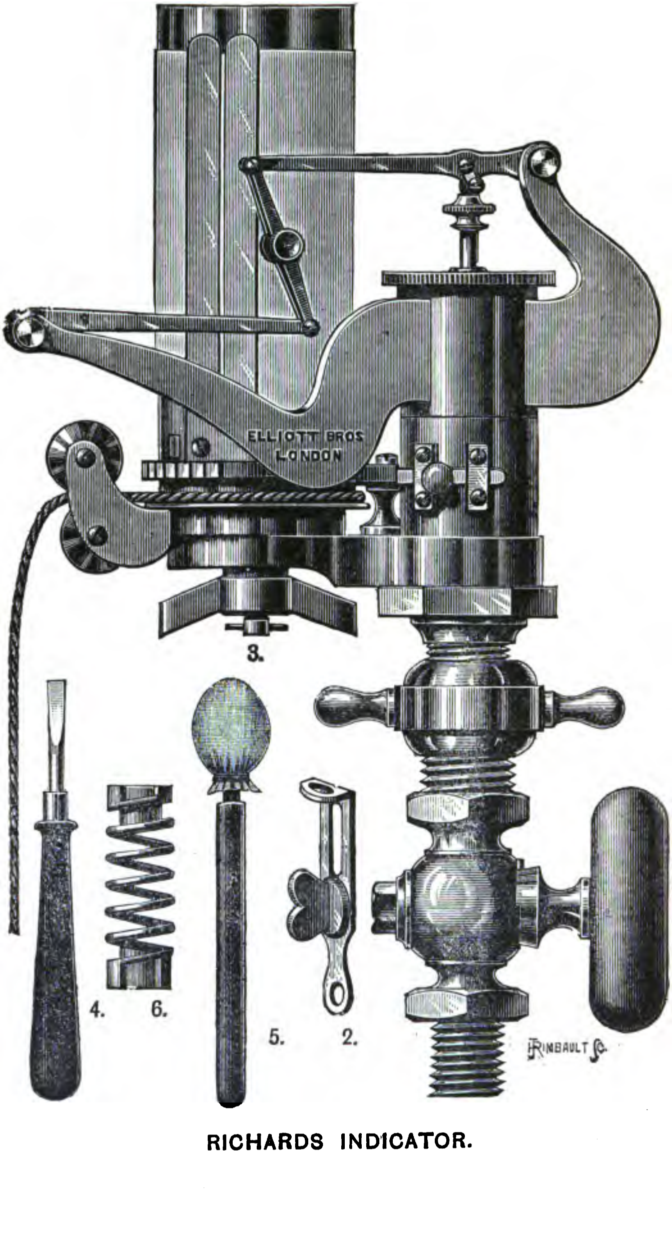
THE RICHARDS

STEAM-ENGINE INDICATOR

Frontispiece.



RICHARDS INDICATOR.

A TREATISE

ON

THE RICHARDS STEAM-ENGINE INDICATOR

(MANUFACTURED BY ELLIOTT BROS., LONDON)

AND THE

DEVELOPMENT AND APPLICATION OF FORCE IN THE STEAM-ENGINE

BY

CHARLES T. PORTER

FIFTH EDITION, REVISED AND ENLARGED

London:
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1894

PREFACE.

ANOTHER Edition of this Treatise having been called for, we have thought it best to reproduce the work (with necessary corrections) as re-written by Mr. CHARLES T. PORTER a few years ago.

A drawing of the original Indicator is shown in this work, also one of the Indicators up to date, several constructional improvements and additions having since been made to the instrument, and we shall be glad at any time to furnish full descriptive particulars and prices to any one who may require them.

At the end of the book we show the latest "Wayne" Indicator, and give a short description of same.

ELLIOTT BROTHERS,
Manufacturers.

Sole Address:
101 & 102 ST. MARTIN'S LANE,
LONDON, W.C.

ADVERTISEMENT.

In introducing Richards' Improved Steam-Engine Indicator, we desire to call the attention of the numerous class who, as constructors, managers, or owners, are interested in the steam-engine, to the advantages which it possesses. In the following pages all necessary information is furnished concerning the instrument and its application, and such instruction is given to those who are not already skilled in the use of the Indicator, as will enable them to employ it to the best advantage.

The Indicator was invented by Watt. For some time it was kept by him a secret, but became known before his death, and to its use, now quite general, we are more indebted than to anything else for the degree of excellence which the steam-engine has attained. The employment of more rapid velocities of piston, with higher pressures of steam, and higher grades of expansion, which has become so extensive, and promises ultimately to be universal, has greatly increased the importance of the Indicator, since this is the only means as yet known, by which the engineer can render himself familiar with the action of steam under these new conditions. Unfortunately every form of this instrument has hitherto failed in its application to engines of this class. The long and tremulous spring used in them was put in a state of violent oscillation by the momentum of the piston and its attached parts, and the result was a serrated figure, from which but little information could be extracted; so that, after a time, attempts to employ the Indicator in this important and rapidly enlarging field were mostly abandoned.

Under these circumstances, the appearance at the Great

Exhibition of 1862, of the improved form of this instrument, invented by Mr. Charles B. Richards, an engineer at Hartford, Connecticut, U.S., may not improperly be regarded as an event of some importance. The action of this Indicator was found to be quite perfect, under the severest tests to which it could there be subjected, and recently it has been still more thoroughly tried, on an express engine on the London and South-Western Railway, and its performance has more than realised the expectations formed of it. Two instruments, among the first manufactured by us, were employed, with which nearly two hundred diagrams were taken, on a trip to Southampton and back, at pressures varying from 80 lbs. to 130 lbs., at rates of motion varying from the slowest up to 260 revolutions per minute, giving a speed of 55 miles per hour, and at all points of cut-off; and they were found uniformly to work with the same steadiness at the highest velocity as at the lowest, and at the earliest point of cut-off as at the latest. Copies of a few of the diagrams are here given.

Owing to the greatly increased speed and high pressures at which engines are now worked, it has been found necessary to introduce a new form of Indicator to meet this want, and we have succeeded in producing an efficient instrument to indicate correctly at speeds up to 1000 per minute, without increasing the friction or inertia of the moving parts. (See description at end of this edition.)

We have only to add, that no pains have been spared to attain, in the manufacture of these instruments, the highest degree of accuracy and excellence, and that, if the directions here given are attended to, their indications may be implicitly relied on.

ELLIOTT BROTHERS,
Sole Manufacturers.

101 & 102 St. Martin's Lane, London.
1894.

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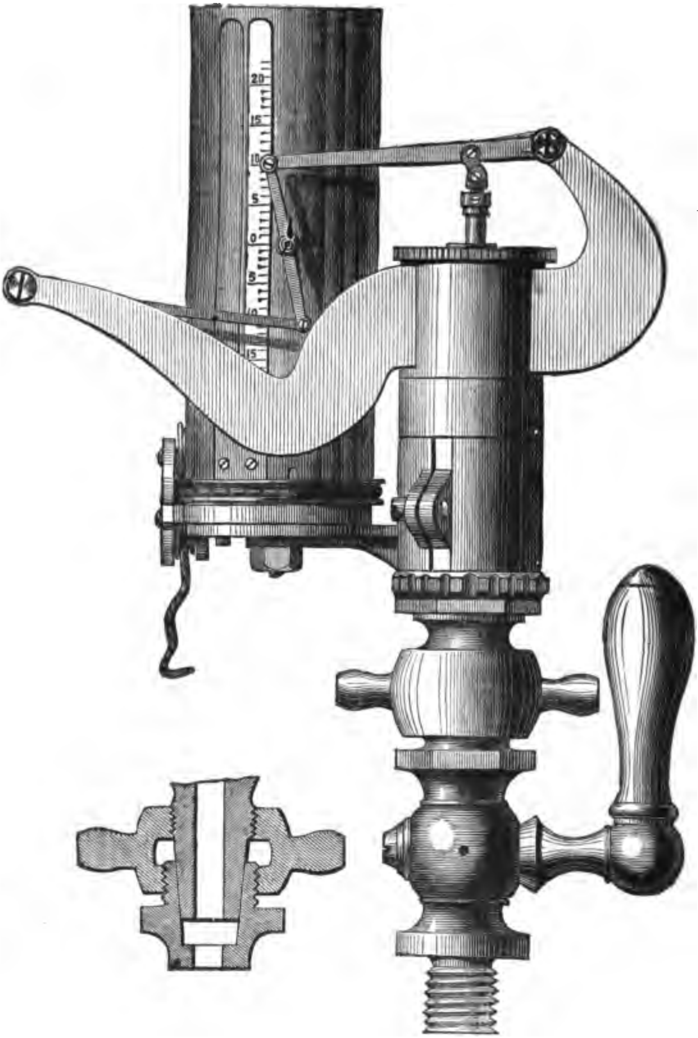


RICHARDS INDICATOR AS ORIGINALLY MADE.

THE

RICHARDS IMPROVED

STEAM-ENGINE INDICATOR.

PART FIRST.

SECTION I.

THE NATURE AND USE OF THE INDICATOR.

THE Steam-Engine Indicator is an instrument designed to show the pressure of steam in the cylinder at each point of the piston's stroke. It does this in the following manner. A pencil, moving up and down with the varying pressure of the steam, draws a line on paper, which has at the same time a motion forward and backward, coincident with that of the piston. The paper is placed on a drum, which is caused, while the piston is advancing, to make about three-quarters of a revolution, by means of a cord connected with a suitable part of the engine, and is brought, while the piston is receding, back to its first position by the reaction of a spring.

The pencil is attached to a small piston of carefully determined area, which moves without sensible friction, and the motion of which is resisted by a spring of known elastic force. In adapting the Indicator to English measurements, the area of the piston is made equal to a certain fractional part of a square inch; and the spring by which its motion is resisted is so proportioned in strength, that a change of pressure of one pound on the square inch shall cause the pencil to move through a certain fractional part of a linear inch.

The pressure of the atmosphere is always on the upper side

of the piston, and when the communication with the cylinder of the engine is closed it is on the under side also; and if then the motionless pencil be applied to the moving paper, it will draw on it a straight line, which is called the atmospheric line. When the communication is opened between the under side of the piston and one end of the cylinder of the engine, the piston will be forced upward by the pressure of the steam, or downward by that of the atmosphere, as the one or the other preponderates; and if now the pencil be applied to the moving paper it will describe, during one revolution of the engine, a figure, each point in the outline of which will show, by its distance above or below the atmospheric line, the pressure in that end of the cylinder when the piston was at the corresponding point of its forward or return stroke.

The diagram thus described shows on inspection the following particulars, viz.: what proportion of the boiler-pressure is obtained in the cylinder—how early in the stroke the highest pressure is reached—how well it is maintained—at what point, and at what pressure, the steam is cut off—whether it is cut off sharply, or in what degree it is wire-drawn—at what point, and at what pressure, it is released—in a non-condensing engine, whether it is freely discharged, or what proportion of it remains to exert a counter-pressure additional to that of the atmosphere —in a condensing engine, the amount of the vacuum, and how quickly or gradually it is obtained—and in both classes of engines, whether, before the commencement of the stroke, there is any compression of the vapour remaining in the cylinder, and if so, at what point it commences, and to how high a pressure it rises. From the diagram, the mean pressure exerted during the stroke to produce and to resist the motion of the piston may be ascertained, and thus the engineer may come to know accurately the amount of power required to overcome the whole aggregate resistance on the engine; and also, by taking separate diagrams for each, the power required by each of the several resistances or classes of resistance separately. He may endeavour also to ascertain the *causes* of the various features presented in the diagram, and thus to learn the effect produced by this or that form or arrangement of the parts, and to detect any imperfection in their construction or action.

It must be borne in mind, that the Indicator shows only the pressure at each point of the stroke; to represent this faithfully

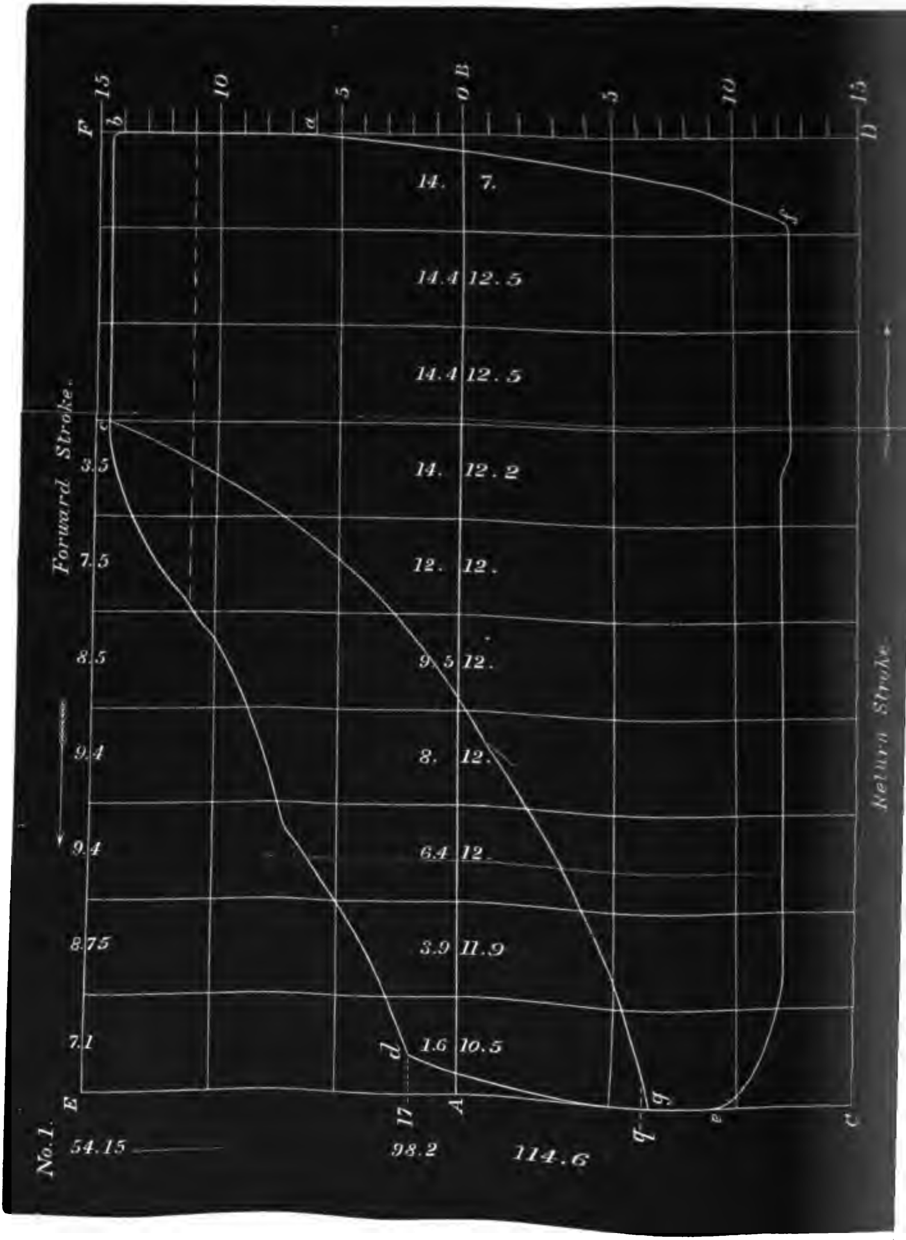
is its sole office. It tells nothing about the causes which have determined the form of the figure which it describes. The engineer concludes what these are, as the result of a process of reasoning, and this is the point where errors are liable to be committed. Conclusions which seem obvious sometimes turn out to have been wrong, and the ability to form an accurate judgment, as to the causes of the peculiarities presented in the diagram, is one of the highest attainments of an engineer.

The variety of diagrams given by different engines, and by the same engine under different circumstances, is endless; and there is perhaps nothing more instructive to the student of engineering, as there is nothing more interesting to the accomplished engineer, than their careful and comprehensive study, with a knowledge of the modifying circumstances under which each one was taken. Lines which at first appeared meaningless become full of meaning; that which then scarcely arrested his attention comes to possess an absorbing interest; he becomes acquainted with the innumerable variety of vicious forms, and learns the points and degrees, as well as the causes, of their departure from the single perfect form; he becomes familiar with the effects produced by different constructions and movements of parts, and competent to judge correctly as to the performance of engines, and to advise concerning changes by which it may be improved; he ceases to be a mere imitator of material shapes, and learns to strive after the highest excellence, and at the same time to comprehend its conditions. No one at the present day can claim to be a mechanical engineer who has not become familiar with the use of the Indicator, and skilful in turning to practical advantage the varied information which it furnishes.

Besides its employment on the cylinder, we must not omit, in this brief summary of the uses of the Indicator, to note its application to boilers, to the steam and exhaust-chambers, and to the condenser and air-pump, and also to vessels of whatever nature in which either elastic or inelastic fluids are under varying pressures. These uses of the instrument will be considered in their appropriate place.

Diagram No. 1, on the next page, was taken in the year 1861, from one of the engines of a then well-known steamship, the *Africa*, of the Cunard Line; and is introduced here to illustrate the application of the Indicator, as above described.

It reminds us also of the great advance which has been made in marine engineering since the day when such diagrams were


Forward Stroke.
Return Stroke.
No. 1.
54.15
98.2
114.6
14. 7.
14.4 12.5
14.4 12.5
14. 12.2
12. 12.
9.5 12.
8. 12.
6.4 12.
3.9 11.9
1.6 10.5
3.5
7.5
8.5
9.4
9.4
8.75
7.1
17

regarded as admirable. Within how short a time have the class of engines that gave them become, like the paddle-wheels which they revolved in ocean navigation, not only antiquated but almost forgotten!

The several lines on the diagram will be designated in this treatise as follows, reference being had to the foregoing figure:—

I. Base Lines.

A B is the *atmospheric line*, drawn in the manner above explained.

C D is the *line of perfect vacuum*. This cannot be drawn by the Indicator, but must be drawn by hand, parallel with the atmospheric line, and at the proper distance below it to represent the pressure of the atmosphere, as shown by the barometer, according to the scale of the diagram. When the actual pressure is not known, it is to be assumed at 14·7 lbs. on the square inch, corresponding almost exactly with 30 inches of mercury, which is about the average pressure at the level of the sea. The barometric column falls one one-hundredth of its height for every 262 feet of elevation above the sea level.

E F is the *line of boiler pressure*. This can be drawn by the Indicator set upon the cylinder only when the engine is at rest, being prevented from moving, while an equilibrium of pressure is established between the boiler and cylinder. It may, however, be drawn when the engine is in motion, by placing the Indicator on the boiler, or, when the flow of steam into the cylinder is stopped early in the stroke by cutting off, by placing it on the steam-chest. Or, the gauge having been proved by the Indicator, this line may then be drawn in by hand, parallel with the atmospheric line, according to its indication.

II. Divisions of the Outline drawn by the Instrument during a Revolution of the Engine.

These are as follows:—

a b, the admission line;	*d e, the exhaust line;*
b c, the steam line;	*e f, the line of counter-pressure;* and
c d, the expansion curve;	*f a, the compression line.*

Of these divisions, the first four are drawn during the forward stroke of the piston, and while it is at, or very close to,

the termination of its stroke, and the last two are drawn during the return stroke.

The admission line shows the rise of pressure, from the point at which the valve commences to open the port admitting steam from the chest to the cylinder, until the highest pressure is reached. In some engines this will not be completed until the piston has advanced more or less in its stroke.

The steam line marks the interval during which the steam, after having reached its full density in the cylinder, continues to flow into it in order to maintain this density and corresponding expansive force more or less completely behind the moving piston. It extends from the termination of the admission line to the point at which the return movement of the valve has completely covered the port, which is called the point of suppression, or cut-off. Ordinarily this point cannot be identified on the diagram, because the gradual contraction of the port causes a fall of pressure in the cylinder, sometimes very considerable, before the closure is completed. Such was, in a moderate degree, the case in this instance, where the port was closed at some unknown point considerably beyond *c*.

The expansion curve represents the fall in the density or pressure of the steam confined in the cylinder, consequent upon the enlargement of its volume by the advance of the piston, from the closing of the admission-port to the opening of the passage for its release or discharge. A theoretical expansion curve has also been drawn in by hand on this diagram, from the point *c*, taken at three-eighths of the stroke, which is considerably before the actual closing of the port. This is for use hereafter.

The exhaust line shows the fall of pressure caused by the release or discharge of the steam, from the point at which the valve commences to open the exhaust passage, until the piston has commenced its return stroke. Sometimes the point of release cannot be identified on the diagram, but on this one it is sharply indicated, and the fall of pressure is rapid.

The line of counter-pressure extends from the commencement of the return stroke to the point at which the exhaust passage is closed. This latter point is here, like the point of release, exactly shown; as we should expect that it would be, since the closure is produced by the reverse movement of the valve, passing the same point with the same velocity as when it effected the release.

The compression line extends from the latter point to the end of the return stroke. It shows the increase in the density, or rise of pressure, of the confined steam, from the diminution of its volume caused by the completion of the return stroke, at the end of which it has been compressed into the space afforded by the clearance and passages.

The scale of this diagram is marked at the commencement. The diagram is divided into ten equal parts, by lines drawn perpendicular to the atmospheric line, and lines parallel with this line are also drawn at intervals of five pounds' pressure. The object of these is to enable the engineer to observe more accurately the features of the diagram, and to ascertain the mean pressure exerted during the stroke, the mode of doing which will be explained hereafter.

From an examination of this diagram, we conclude that the exhaust-port was covered at the point *f* of the return stroke, and the vapour remaining in the cylinder was then compressed by the advance of the piston to a density, at the commencement of the forward stroke, of about five pounds above the atmosphere. The port was then opened for admission, and the pressure instantly rose to fourteen and a half pounds above the atmosphere. The port being opened wider and wider, this pressure was maintained behind the piston to a point beyond *c*, at which it began to fall, at first very slowly, from the gradual closing of the port by the return movement of the valve. The point at which the port was covered cannot be identified. It was considerably beyond the point *c*, and the steam line continues to the point of cut-off, however much the pressure may fall before that point is reached. At the point *d*, the pressure had fallen by expansion to two pounds above the atmosphere. Here the valve began to open communication with the condenser, and before the piston commenced its return stroke the pressure on this side of it fell to nearly ten pounds below the atmosphere, and almost immediately after a vacuum of twelve pounds was formed; and when the return stroke was two-thirds accomplished, the counter-pressure suddenly fell half a pound lower, and this vacuum was maintained until the exhaust-port was closed at the point *f*. We shall refer to this diagram again, when on the subjects of calculating the power of the engine from the diagram, and of working steam expansively; but will point out here an excellent illustration which it affords of the practical value of the Indicator, provided

the intelligence and care to observe and follow its indications are found in the engineer.

A singular rise of pressure will be noticed in the expansion curve, and also on the line of counter-pressure a sudden improvement in the vacuum, which has already been mentioned. These have a cause, and no competent engineer would rest till he had found it. They take place, it will be seen, at points about equally distant from the terminations of the forward and the return strokes. It is therefore probable, though not certain, that their cause is one and the same. It would seem as if nothing but a leakage from the chest into the cylinder could occasion a rise of pressure like this during the expansion; and if so, then it was suddenly stopped at the point indicated in the diagram. If this leak occasioned also the loss of vacuum shown, and the stoppage of it produced the improvement, then it must have been a big one. There was reason to apprehend here the existence of a serious defect, involving a great waste of steam, which, but for the Indicator, might never be suspected.

The rise of the compression line, also, is more rapid than could have been produced by compression of the confined steam. This must have been supplemented by leakage from the chest. We shall see by-and-by how this is to be detected.

The necessity for the employment of the Indicator, if anything except the wildest guess-work respecting the condition, or performance, or power of an engine, is expected, has its frequent illustration in the experience of every engineer accustomed to its use.

The writer was at one time applied to to supply a hundred horse-power engine for a certain manufactory. "Are you quite sure that you will require an engine of that size?" "O yes: our consulting engineer, in whose judgment we place entire confidence, [who, by the way, was the patentee of the boiler used], tells us that we are using fifty horse-powers now [of course, with admirable economy], and we are expecting to double our capacity, and so shall need a hundred horse-powers in the new engine." "Has your engine been indicated?" "No." "How has your engineer determined the power?" "That we know nothing about." We proposed to measure the power they were then using. The consulting engineer was indignant and supercilious; but, since we would be answerable for its economical performance, we declined to furnish the

engine unless we could ascertain for ourselves the power required.

The Indicator, when applied, showed 90 lbs. pressure in the boiler, to be reduced, by means of a small and crooked pipe, to 45 lbs. in the cylinder, the back-pressure of the atmosphere to be increased by 7 lbs. additional resistance of the steam, and the total power exerted by the engine to be 25 horse-powers, of which 14 were exerted to drive the shafting; so that when the enlargement of the works should be completed, no more than 40 horse-powers would be required.

An engine was running to furnish auxiliary power in a mill where an aggregate water-power was used of 700 or 800 horse. It was represented and believed to be exerting 200 horse-powers, and the economy was quite astonishing. The application of the Indicator confounded all parties by showing only 60 horse-powers exerted by that engine.

What foolish misadaptations are sometimes made for the want of the information the Indicator will give! The writer saw, some years ago, in Manchester, a beam-engine of respectable size, with air-pump to correspond, supplied to exhaust the pipes of the atmospheric telegraph. It was found to be almost impossible to carry a sufficiently low pressure, or to run at sufficiently slow a speed. An engine which two men could have lifted would have done the work, and done it better.

The use of the Indicator makes the engineer sure and safe in his calculations of power. We had to supply a pair of small engines to drive two electro-magnetic machines for Professor Holmes' Light-house at the Paris Exposition in 1867. The requirement was that either engine should drive either or both machines. We went up to London, and ascertained by application of the Indicator the power required to drive one similar machine, and proportioned the engines accordingly, knowing the steam-pressure on which we could rely. When a start was to be made at Paris, in the presence of a large company, at first nothing would move; with every exertion both engines could not even be made to drive one machine, except at a snail's pace. The Superintendent of the British Mechanical Department was rash enough to declare—"There has been a great blunder made here in providing the power." Knowing that the trouble was not in the engines, we requested both machines to be disconnected; when, as we expected, both engines, with the steam

following the piston as far as possible, could not drive the intermediate gearing. On examination, this was found to be so defective that it could not be run. Proper gearing was procured to replace it, and on the next trial one engine, when cutting off the steam at one-sixth of the stroke, proved exactly suited to drive the two machines, as intended.

SECTION II.

OF TRUTH IN THE DIAGRAM.

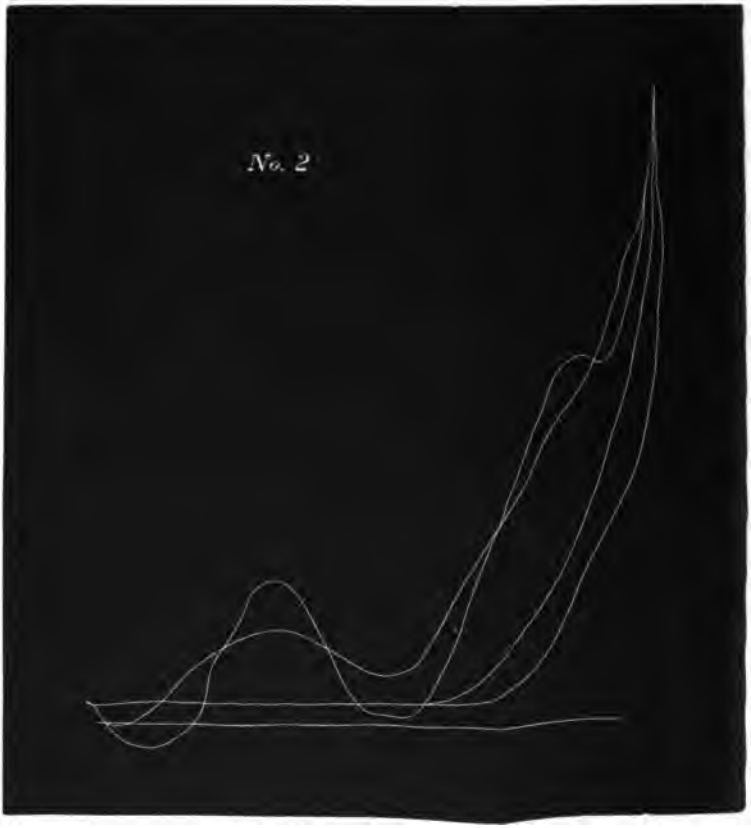
It is of course of the first importance that the diagram given by the Indicator shall be true. Causes of error appear at every point, and the degree of falsity arising therefrom increases greatly with an increase in the rate of revolution of the engine. It is not possible to be too critical in using the Indicator especially at high speeds; the errors we are not conscious of are the ones sure to mislead us.

The conditions of a correct diagram are—1st, that the movements of the paper shall coincide exactly with those of the piston; and 2nd, that the movements of the pencil shall *simultaneously* and precisely represent the changes of pressure in that end of the cylinder to which the Indicator is attached.

1st. *Errors in the motion of the paper.*—The common errors in communicating motion to the paper are of two kinds—those which arise out of the movements employed, and those which, when the movements are correct, are occasioned by a high velocity of the parts: but with proper care these may all be avoided. We shall mention them in detail presently, in connection with instructions for applying the Indicator.

2nd. *Errors in the motion of the pencil.*—These are of a more serious nature. The spring may be accurate, but, in all the forms of the Indicator hitherto in use, its unavoidable length and weakness, and its weight, joined to that of the piston and other attached parts, and the distance through which these must move, in order that the indications may be on a scale of sufficient magnitude, render it impossible to obtain from engines which run at any considerable speed diagrams which can make any claim to accuracy. The two following diagrams, Nos. 2 and 3, afford good illustrations of this. They are fair average samples of a large number taken in February, 1856, by the late Daniel Kinear Clark, from the locomotive "Canute," on the London and South-Western Railway, with an Indicator of the best construction then known, and which had been expressly prepared for the purpose. The speed and pressure are not

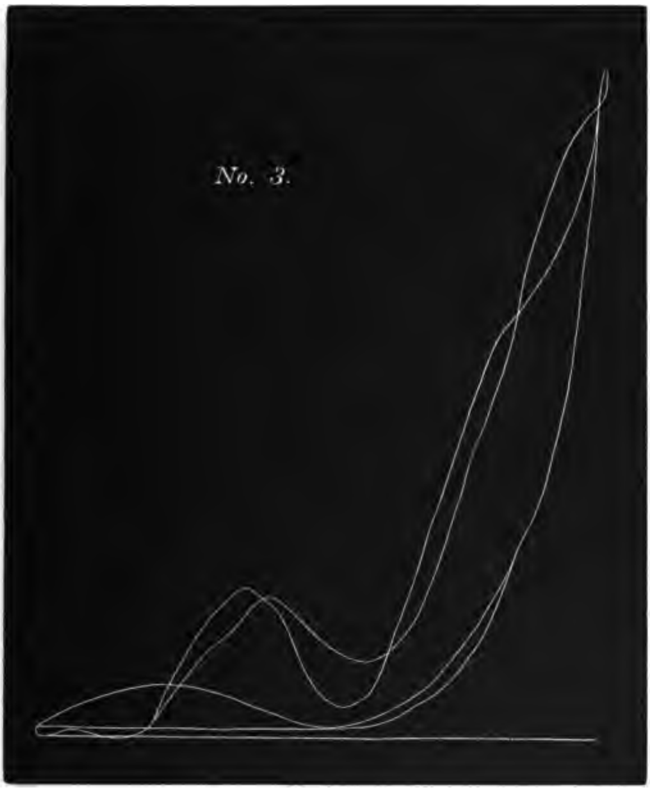
noted on the originals, which are here reproduced exactly. The attempt to conjecture what the true form of these diagrams should be—to learn from them, for example, what proportion of the boiler-pressure was obtained in the cylinder, and how much the pressure fell before reaching the point of cut-off, at the speed of piston employed—points which it is of the highest consequence to ascertain—is clearly hopeless. It is to be



observed also that the pencil does not, in either case, follow the same line during two successive revolutions of the engine, but describes quite different figures.

Until the cause of these vibrations was revealed by the Richards Indicator, the opinion had become quite general among engineers who had most attentively considered the subject, that

they represented actual pulsations of the steam in the cylinder. (*See* Clark's *Railway Machinery*, pp. 72, 73.) The Richards Indicator was applied to the same engines, and lo! there were none. This is a remarkable illustration of how experience and the greatest acumen may be misled, and of the method of varied experiment, by which alone the proper understanding of any operation of nature is to be got.

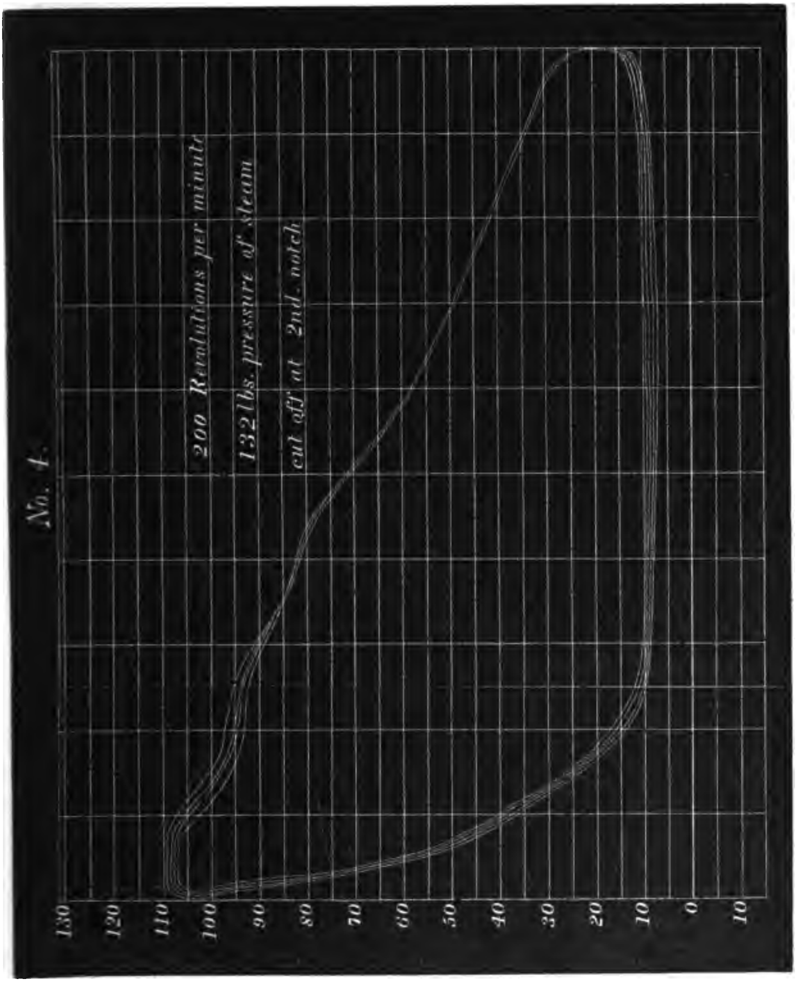


But other errors, not so obvious, arise from the same causes. The long spring cannot be fully or nearly compressed without bending, when one side of it presses against the surface of the case in which it is confined, and the piston is forced against the opposite side of the cylinder, and from both the bending and the friction errors in the diagram necessarily result. The distance through which the piston moves, moreover, makes it

impossible that changes of pressure shall be simultaneously indicated; the piston of the engine must travel while the action is taking place, and thus the diagram shows the changes of pressure later, or more gradually than the fact. At high speeds this becomes a serious error. In addition to this, the distance through which the piston of the Indicator moves occasions a fall of pressure between the cylinder of the engine and that of the Indicator, which also becomes of consequence at high velocities. For the purpose of diminishing this most obviously bad action of Indicators, it has been usual to make them with small areas of piston—generally one quarter, sometimes even one-eighth of a square inch; but it is questionable if this does not rather increase the number and magnitude of the errors. The spring, the length of which cannot be reduced, becomes exceedingly slender, very slight friction checks its play, the atmospheric line cannot be drawn with precision, and the reaction on fall of pressure is feeble.

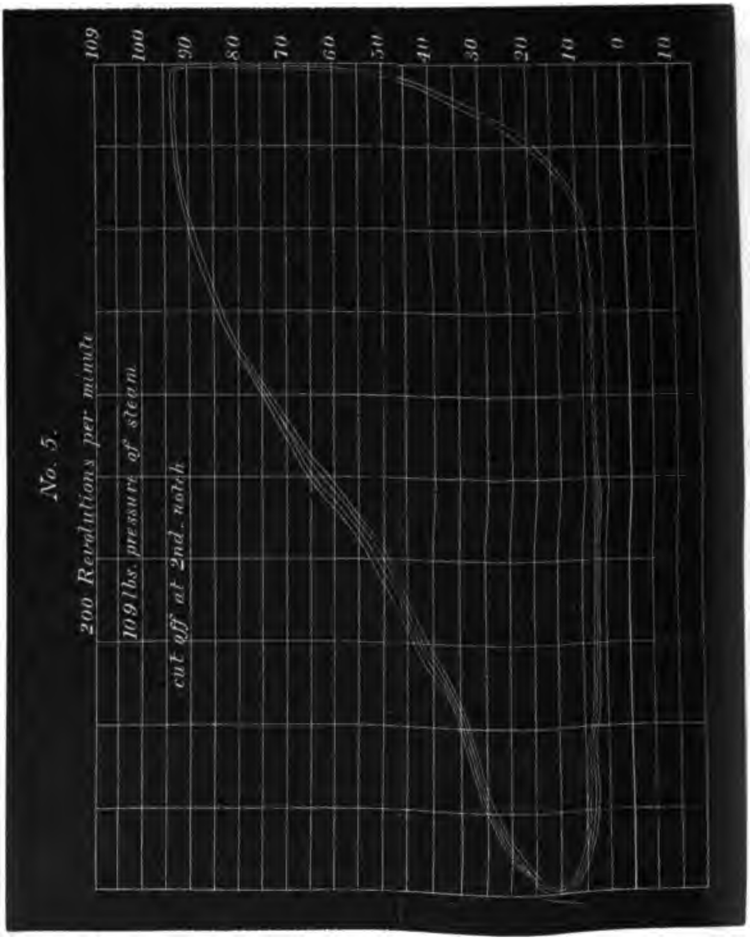
SECTION III.

DESCRIPTION OF THE RICHARDS INDICATOR.



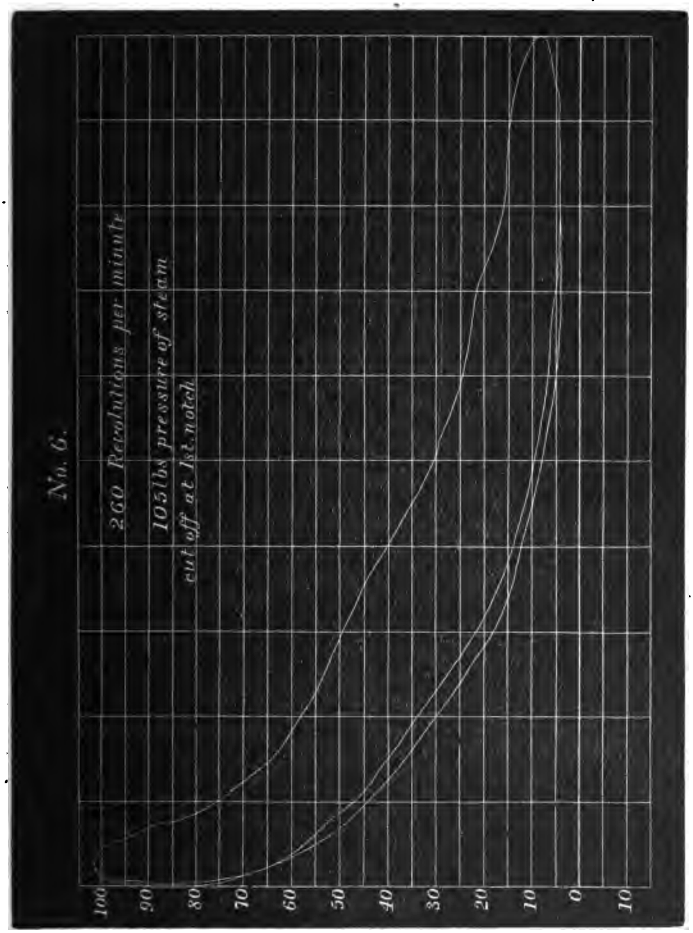
This Indicator is constructed on a plan by which it is found that the defects pointed out in the preceding section are quite

avoided, and correct diagrams are obtained under all circumstances. The principal distinguishing features of this instrument are, a short and strong spring, a short motion of piston, and light reciprocating parts, combined with a considerable



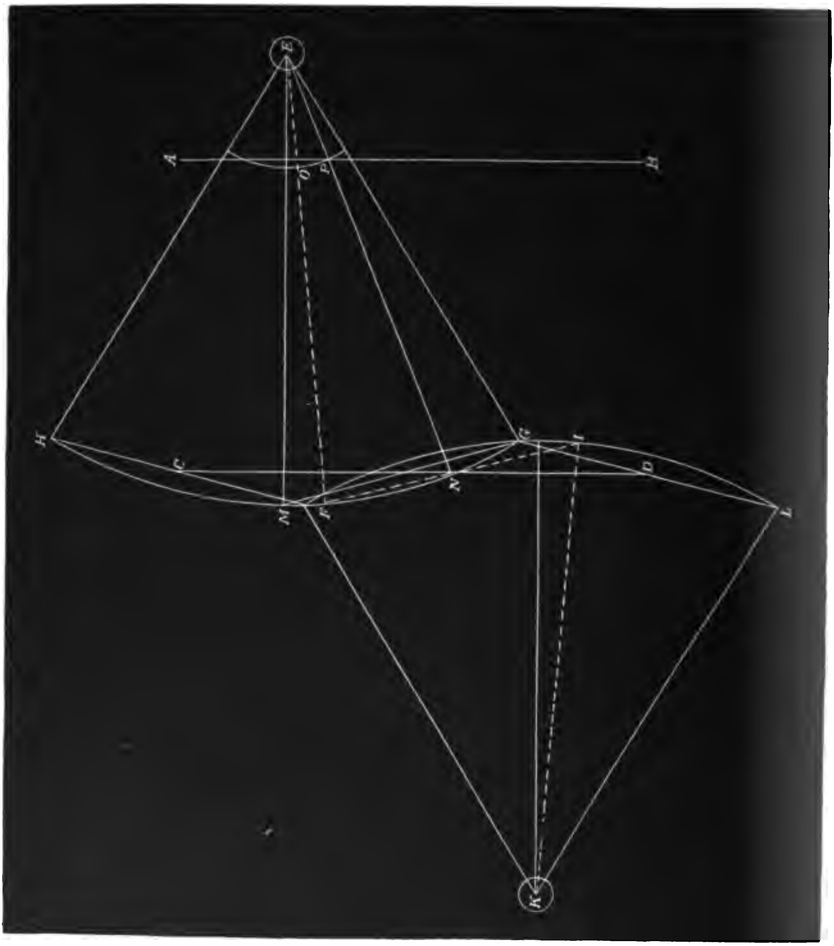
area of cylinder, and an arrangement of levers and a parallel motion, for multiplying the motion of the piston in such a manner that the diagram is described in the usual way and of the ordinary size. The proportion between the motion of the

piston and that of the pencil is a matter of discretion; that which has been adopted is 1 to 4, and the steadiness with which the indication is drawn by these instruments, even at the highest speeds of piston, leaves nothing to be desired.



The diagrams numbered 4, 5, 6, 7, are fair samples of a large number taken with this Indicator from the locomotive "Eagle," on the London and South-Western Railway, in April, 1863. In three of them the pencil was held to the paper during a

number of revolutions; in diagram No. 6 it passed over the paper only once and a half. They are introduced here to show the correct action of the instrument: we shall hereafter have



occasion to consider them also as illustrations of working steam expansively.

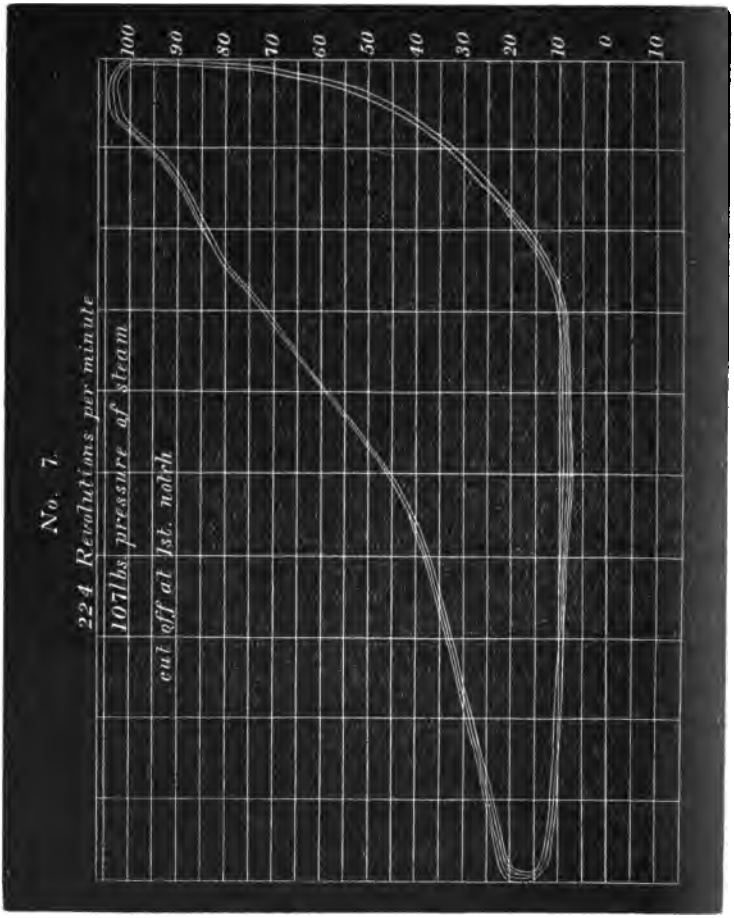
In respect to the ability of these Indicators to give diagrams

which shall be *perfectly accurate*, free from occult as well as from palpable errors, it is to be observed that the spring moves without any tendency to bend, and the motion of the piston, and the length of cylinder to be filled with steam as the piston rises, is one-fourth of that in the ordinary Indicator. It is assumed also that if the motion could be frictionless, then the approach to simultaneousness in the action of an Indicator would be in a direct ratio to the strength of the counteracting forces existing in the pressure of the steam on the one side of the piston, and the resistance of the spring on the other, and in an inverse ratio to the distance through which the piston has to move upon any given disturbance of their equilibrium. But, moreover, the motion cannot be absolutely frictionless; and if the friction should be equal in two Indicators of different strokes, then the resistance from it would be, in each one, directly as the length of its stroke; and if the resistance from friction should be equal in two Indicators having different areas of piston, then its effect on the diagrams given by them would be in an inverse ratio to the areas of the pistons. In every view which can be taken, it is evident that the features embodied in this indicator—namely, a strong spring, short motion of piston, and light moving parts, combined with a reasonably large area of piston—are essential for the attainment of truth in the diagram.

The motion of the pencil.—A correct motion of the piston having been obtained, the question presents itself:—How shall the motion of the pencil be made to represent this, on any magnified scale, accurately at every point? This problem has been very happily solved by the inventor of this instrument, by an application of the parallel motion. The opposite diagram represents the movements of the light steel arms employed for this purpose.

A B is the line of motion of the piston, extended for convenience of representation, and C D that of the pencil. The dotted line, E F, represents a lever, which turns on a fulcrum at E, and vibrates in the arc G H. It is connected by the link F I with the arm I K, which vibrates about the point K, in the arc L M. These are so placed that the centre, N, of the link F I, at which point the pencil is fixed, moves in the line C D, parallel with the line A B. The lever, E F, is connected at the point O with the piston-rod, by the link O P, parallel with the link

F I, and since E O : E F :: O P : F N, it follows that the links O P and F I, if parallel in one position of the levers, will be so also in every position; a straight line will always connect the points E, P, and N, and the motion of the point N in the



line C D will perfectly represent that of the point P in the line A B.

General construction of the Indicator.—The parallel motion is

made as compact as possible. For this purpose a lever of the third order is employed to multiply the motion, and the extremities of the line drawn by the pencil are permitted to have a slight curvature, as shown in the diagram, which considerably reduces the length of the rods, and does not affect the usefulness of the instrument, the curvature at the lower end being below any attainable vacuum, while the extremity of the scale above is rarely employed.

The Indicators are made of a uniform size, the area of the cylinder is one-half of a square inch, its diameter being ·7979 of an inch. The piston is not fitted quite steam-tight, but is permitted to leak a little; this renders its action more nearly frictionless, and does not at all affect the pressure on either side of it. It must be distinctly understood that leakage, unless it be sufficient to add to the atmospheric pressure above the piston, cannot affect the accuracy of the indications. Absurd devices have even been patented for keeping the piston tight, while the one thing needful is absolute freedom of motion.

The motion of the piston is $\frac{25}{32}$ of an inch, and the motion of the pencil, or extreme height of the diagram, is $3\frac{1}{8}$ inches. The paper cylinder is 2 inches in diameter, and the length of the diagram may be $5\frac{1}{4}$ inches, if this extent of motion is given to the cord. The best length is about 4·5 inches. The diagram is drawn by a pointed brass wire on metallic paper. This is a great improvement over the pencil; the point lasts a long time, cannot be broken off, and is readily sharpened, and the diagram is indelible. The steam-passage has two or three times the area usually given to it. The stem of the Indicator is conical, and fits in a corresponding seat in the stop-cock, where it is hold by a peculiar coupling, shown in section in the accompanying cut of the Indicator. This arrangement permits the Indicator to be turned round, so as to stand in any desired position, when, the coupling being turned forward, the difference in the pitch of the screws draws the cone firmly into its seat; and when the coupling is turned backward, the cone is by the same means started from its seat. The leading pulleys may be turned, by some pressure, to give any desired direction to the cord, and will remain where they are set. By these means the Indicator can be readily attached in almost any situation. When required for use on oscillating cylinders, the Indicator is

furnished with an attachment which prevents the pencil from being affected by the motion of the cylinder.

The springs.—In order to adapt this Indicator for use on engines of every class, springs are made for it to ten different scales, as follows:—

No. 1, $\frac{1}{8}$-in. motion, shows 1 lb. pressure on the sq. in. Indicates from					− 15 to	+ 10
„ 2, $\frac{1}{12}$	„	„	„	„	− 15 „	+ 22·5
„ 3, $\frac{1}{16}$	„	„	„	„	− 15 „	+ 35
„ 4, $\frac{1}{20}$	„	„	„	„	− 15 „	+ 47
„ 5, $\frac{1}{24}$	„	„	„	„	− 15 „	+ 60
„ 6, $\frac{1}{30}$	„	„	„	Atmosphere	to	+ 80
„ 7, $\frac{1}{32}$	„	„	„	„	„	+ 100
„ 8, $\frac{1}{40}$	„	„	„	„	„	+ 125
„ 9, $\frac{1}{48}$	„	„	„	„	„	+ 150
„ 10, $\frac{1}{56}$	„	„	„	„	„	+ 175

Springs are made also to any other scale required.

Each of these springs will fit every instrument alike, and they can be sent with the boxwood scales by post to any address, and the spring can be changed in the instrument by any one in a few minutes, in the manner explained hereafter in the "Directions for the care of the Instrument." Most of the scales are multiples of 8, and the common rule will measure the diagrams, if the proper scale is not at hand. It will be observed that the five higher scales do not indicate the vacuum. These are so made for the following reasons. The far greater number of engines which work steam at high pressures do not condense, and moreover, at these pressures, the scale of the indication necessarily becomes small, while it is always desirable to show the vacuum on a large scale. Spring No. 1 may be employed to indicate the vacuum, in engines which work steam at high pressures and with condensation. It can be readily substituted in the Indicator, and the diagram given by it will be on a satisfactory scale. It is provided with a stop, which prevents it from being compressed too much, so that a high pressure of steam will not injure it. Moreover, the vacuum being omitted from the scales which go above 60 lbs., the entire range of the pencil is available for the pressures above the atmosphere, which may therefore be shown on a larger scale. Springs indicating pressures above 60 lbs. are supplied, however, to indicate the vacuum also, when so ordered.

The springs are tested with a highly sensitive apparatus, designed expressly for the purpose, and are corrected for a temperature of 212°, which is the temperature to which they will be exposed under almost all circumstances when in use, and at which their accuracy is guaranteed.

SECTION IV.

PRACTICAL DIRECTIONS FOR APPLYING AND TAKING CARE OF THE INDICATOR.

I. Of Attaching the Indicator.

When it is practicable, diagrams should be taken from each end of the cylinder. The assumption commonly made, that if the valves are set equal the diagram from one end will be like that from the other, will be shown by this instrument to be erroneous. This is owing to the difference in the speed of the piston at the opposite ends of the cylinder, which is, at the outer end of a direct-acting engine, from 35 per cent. to 66 per cent. greater than at the crank end, the difference varying according to the degree of angular vibration of the connecting rod. In side-lever or beam-engines, these proportions are reversed, and the speed of the piston is greater at the upper end of the cylinder. Often also there is a difference in the lengths of the thoroughfares, and in the lead, or the amount of opening, or the point of closing; and many times the valves are supposed to be correctly set, when this Indicator will show that they are not. These, and other causes, will make a difference in the diagrams obtained from the opposite sides of the piston.

One use of the Indicator is in fact to show whether or not the diagrams from opposite ends of the cylinder *are* alike.

Pipes to be avoided.—The Indicator should be fixed close to the cylinder, especially on engines working at high speeds. If pipes must be used, they should not be smaller than half an inch in diameter, and five-eighths in the bends, and as short and direct as possible. Any engineer can satisfy himself with this instrument that each inch of pipe occasions a perceptible fall of pressure between the engine and the Indicator, varying according to its size and number of bends and the speed of the piston. Diagrams have been known to show, from this cause alone, 40 per cent. less pressure than was actually in the cylinder. Probably the diagrams taken from engines, generally, show in nine cases out of ten the pressure, or more often

the lead, of the steam, or direction of the admission line, untruly, from the incorrect manner in which the instrument is attached.

Where to connect the Indicator.—On vertical cylinders, for the upper end, the Indicator-cock is usually screwed into the cover. Sometimes it is attached where the oil-cup is set, this being removed for the purpose. For the lower end, it is necessary to drill into the side of the cylinder, at a convenient point in the space between the cylinder bottom and the piston, when on the centre, and screw in a short bent pipe, with a socket on the end to receive the Indicator-cock. The Indicator can be used in a horizontal position, but it will be found much more convenient to put in a bent pipe, and set it vertical. Sometimes it will be necessary to drill in the side of the cylinder at the upper end also, especially in double-cylinder engines having parallel motions, when the Indicator cannot generally be set on the covers. Care must be taken that the piston does not cover the hole when on the centre. No putty is necessary to make these small joints, and it should never be used, as it is liable to get into the instrument. If the screw fits loosely, a few threads of cotton wound round the stem will prevent the escape of steam.

On horizontal engines, the best place for the Indicator is on the top or upper side, at each end; if it cannot be placed there, bent pipes may be screwed into the covers or into the side of the cylinder. In other respects follow the directions given for vertical engines. The Indicator should never be set to communicate with the thoroughfares. The current of steam past the end of the pipe or the hole reduces the pressure in the instrument, and the diagram given is worthless, as any engineer can readily ascertain by making the experiment.

The stop-cock being screwed firmly to its place, screw the Indicator down to its seat, turning it to the most convenient position, and make it fast by turning the coupling; then move the guiding pulleys to their proper position to receive the cord, and the instrument is in readiness for use.

Before attaching the Indicator, open the stop-cocks and let the steam blow through them repeatedly. This should always be done, but it is especially important on new engines. Indicators are sometimes ruined by a mass of core sand and iron filings being blown into them on the very first attempt to use them. The scratched and filthy state into which they are

sometimes allowed to get causes very curious diagrams, which are, of course, a great deal worse than good for nothing. With proper care the instrument may be kept in perfect condition.

II. Of giving Motion to the Paper.

The drum the best means.—The revolution of a drum is probably the most correct as well as convenient method of giving motion to the paper. It may be supposed that a flat slide, worked by positive means, would have a perfectly accurate motion; but, in fact, at high velocities, where alone any trouble is met with, the difficulties involved in its use are more troublesome than those presented by the cylinder. In most cases the connecting-rod must necessarily be somewhat long; it must not tremble, or the line on the paper will be tremulous, and the weight required for stiffness, joined to the weight of the slide causes a momentum, which, if the rod is worked by a vibrating arm, will give to the paper, on each centre, a motion opposite to that of the piston of the engine; and precisely at these points it is of the greatest consequence that the two motions shall coincide.

In the use of the cylinder at any speed, the question of obtaining a positive motion, if there is no elasticity in the cord or the parts to which it is connected, is simply one of proportion between the momentum of the revolving parts and the strength of the spring by which this is resisted. In this Indicator these parts are made as light as possible consistently with other requirements, and the spring is of such strength that they may be reciprocated from 250 to 300 times per minute, without any increase in the length of the diagram; and of course, therefore without any error in the motion. There is no difference in the construction of these Indicators in this respect, it being intended that every instrument shall be applicable to any engine.

From what points to derive the motion.—This may be taken from any part of the engine which has a motion coincident with that of the piston. For a beam-engine, a point on the beam, or beam-centre, or on the parallel-motion rods where these are employed, will give the proper motion; but care must be taken that the cord be led off in the right direction—a requirement which is sometimes overlooked; afterwards its direction of motion may be changed as required.

In some cases it is most convenient to take the motion from a point on the end of the revolving shaft; this is frequently the case on horizontal engines, working at high speeds, because then the motion does not need to be reduced. Exact accuracy cannot be got in this way, however, without employing a moving slide, and connecting it with the pin in the end of the shaft by a rod or cord of such length that its angular vibration shall be the same as that of the connecting-rod. This will be found generally a troublesome matter; and the engineer will probably prefer in most cases to disregard the error resulting from its omission—which is, that the motion of the paper will be more nearly equal at the two ends of the stroke, being slower than that of the piston at the one end, and faster at the other. The crank or pin from which the cord receives its motion must be on its centre, relatively to the direction of the cord, whatever that direction may be, precisely when the crank of the engine is on *its* centre. If this requirement is not carefully attended to, the diagram will be worthless.

Generally, on horizontal engines, the motion of the paper is taken from the cross-head. In an engine-room, a strip of deal may be suspended from the ceiling in such a manner as to permit it to swing backward and forward edgeways by the side of the guides, and motion may be given to it by a pin, secured firmly to the cross-head, and projecting through a slot in the board, in which it should fit nicely to prevent lost time on the centres. The board must hang plumb when the piston is in the middle of its stroke. The cord may be connected to this strip of board at a point sufficiently near to its point of suspension to give the required reduction of motion for the paper, and must be led off in a horizontal direction, and then over one or more pulleys in any required direction to the Indicator. At high speeds, however, pulleys should be avoided, or, when necessarily used, must be firmly held. We have seen them changing their position a full quarter of an inch in every revolution of the engine, from the varying tension of the cord. On portable engines the motion may be obtained in the manner just described, the lever swinging from a pin supported in a standard about two feet in height, set on one of the guide-bars.

On locomotives having outside connections the motion must be taken from the cross-head. It is indispensably necessary to use only a short direct cord, free from elasticity, and connected to a point the motion of which is reduced from that of the cross-

head by positive means. Care must be taken also so to propor tion the parts employed for this purpose, that the point a which the cord is connected shall have a positive motion with out any fling, a matter not by any means free from difficulty a 250 revolutions per minute. A rock-shaft, turning in bushings supported by two angle iron standards, precisely over the mid position of that point of the cross-head from which the motio is derived, affords perhaps the best means of reducing the motion A long arm is worked by the cross-head and a short arm give motion to the cord. The short arm must be keyed in such position that when the piston is in the middle of its stroke i will stand at right angles with the direction of the cord, what ever that may be. The direction of the cord may form an necessary angle with the horizontal line, but must be at righ angles with the rock-shaft.

On locomotives having inside connections, and a single pai of driving-wheels, where it is practicable, it will be found to b the better way to take the motion from a pin set in the end o the axle, and to communicate it by a connecting-rod to a poin convenient for attaching the cord. The parts should be al substantially made; the momentum of the connecting-rod wil be perfectly resisted by the pin.

On oscillating engines, the motion may be taken from the brasses at the end of the piston-rod. If the stroke is long, it is sometimes difficult to reduce this motion to that required for the paper, and in such cases it is necessary to take the motion from an eccentric on the main shaft, to a point as near as possible to the trunnion, and thence to communicate it to the Indicator. In all these connections it is of the first consequence that there be *no* lost time, which will require to be made up on every centre, and will thus cause the paper to stand still while the piston is moving.

It is a difficult matter to take a perfectly correct diagram at high speed. The best test is the length. If the diagram is any longer when taken at high speed than it is at the slowest speed at which the engine can be run, then there is elasticity somewhere, and consequent untruth. Few diagrams from high-speed engines will stand this test; sometimes they are nearly or quite an inch too long. The length can and should be made *the same* as at the slowest speed. After everything else has been made rigid, the cord will give trouble unless it is very short. No matter how hard and fine a line is used, if long its

elasticity will be objectionable. The writer has of late been using brass wire instead. Spring brass wire, No. 27 Stubbs' gauge, has been found to answer admirably. By means of it the motion can be communicated with precision to considerable distances, at 200 or more revolutions of the engine per minute.

Pulleys of different diameters on the same spindle have been used as a means of reducing the motion from that of the cross-head, but we do not recommend them; at high speeds it is very difficult to make them answer, their unavoidable momentum preventing the correct change in the direction of the motion.

The experience of the careful operator will teach him to guard against the various causes of error here mentioned, and others which will arise in the great diversity of situations in which the Indicator is used, and the effects of which are the more mischievous, because often the diagram itself furnishes no means of detecting them. The mathematician will perceive that *perfect* accuracy of motion is attained by only a very few of the methods here suggested. Most of them are only approximately accurate, but they are the best which can be readily employed, and the errors which they involve are too slight to be of practical moment. For the professional engineer, of course directions are unnecessary.

III. How to take a Diagram.

To fix the paper.—For this purpose it is not necessary to remove the paper drum from the instrument. Secure the lower edge of the paper, near the corner, by one spring; then bend the paper round the cylinder, and insert the other corner between the springs. The paper should be long enough to let each end project at least half an inch between the springs. Take the two projecting ends with the thumb and finger, and draw the paper down, taking care that it lies quite smooth and tight, and that the corners come fairly together. The spring used on this Indicator for holding the paper will be found preferable to the hinged clamp. A little practice, with attention to the above directions, will enable any one to fix the paper very readily.

The marking-point should be fine and smooth, so as to draw a fine line, but not cut the paper. It may be made of a brass wire; the best material is gun-metal, which keeps sharp for a long time, and the line made by it is very durable. Lines

drawn by German-silver points are liable to fade. A large-sized common pin, a little blunted, answers for a marking-point very well indeed; a small file and bit of emery-cloth used occasionally will keep the point in order.

To connect the cord.—The Indicator having been attached, and the correct motion obtained for the drum, and the paper fixed, the next thing is to see that the cord is of the proper length to bring the diagram in its right place on the paper—that is, midway between the springs which hold the paper on the drum. In order to connect and disconnect readily, the short cord on the Indicator is furnished with a hook, and at the end of the cord coming from the engine a running loop may be rove in a thin strip of metal, in the manner shown in the following cut, so that it can be readily adjusted to the proper

length, and taken up from time to time, as it may become stretched by use. On high-speed engines it is as well, instead of using this, to adjust the cord and take up the stretching, as it takes place, by tying knots in the cord. If the cord becomes wet and shrinks, the knots may need to be untied, but this rarely happens. All this trouble is avoided by using a wire, as recommended above. A small ring should be attached to the end of the wire to hook on to. It will be found a great convenience also at high speed to have a long hook, out of the way of the fingers. The length of the diagram drawn at high speeds should not exceed four and a half inches, to allow changes in the length of the cord to take place to some extent, without causing the drum to revolve to the limit of its motion in either direction. On the other hand, the diagram should never be drawn shorter than is necessary for this purpose.

To take the diagram.—Everything being in readiness, turn the handle of the stop-cock to a vertical position, and let the piston of the Indicator play for a few moments, while the instrument becomes warmed. Then turn the handle horizontally to the position in which the communication is opened between the under-side of the piston and the atmosphere, hook on the cord, and draw the atmospheric line. Then turn the handle back to its vertical position and take the diagram.

When the handle stands vertical the communication with the cylinder is wide open, and care should be observed that it does stand in that position whenever a diagram is taken, so that this communication shall not be in the least obstructed. The instrument is provided with a stop, hidden in the cylinder-case, to prevent the marking-point from tearing the paper. The arm is to be pressed firmly up to this stop. If the line drawn is faint, the point must be screwed up, and back if the line is too heavy. The elasticity of the parallel arms gives the light pressure required on the paper. As the hand of the operator cannot follow the motions of an oscillating cylinder, it is necessary that the point be held in contact with the paper by a light spring, and instruments to be used on engines of this class are furnished with an attachment for this purpose.

Diagrams should not be taken from an engine until some time after starting, so that the water condensed in warming the cylinder, &c., shall have passed away. Water in the cylinder in excess always distorts the diagram, and sometimes into very singular forms. The drip-cocks should be shut when diagrams are being taken.

As soon as the diagram is taken, unhook the cord; the paper cylinder should not be kept in motion unnecessarily, it only wears out the spring, especially at high velocities. Then remove the paper and minute on the back of it at once as many of the following particulars as you have the means of ascertaining, viz.:—

The date of taking the diagram, and scale of the Indicator.

The engine from which the diagram is taken, which end, and which engine, if one of a pair.

The length of the stroke, the diameter of the cylinder, and the number of double strokes per minute.

The size of the ports, the kind of valve employed, the lap and lead of the valve, and the exhaust-lead.

The amount which the waste room, in clearance and thoroughfares, adds to the length of the cylinder.

The pressure of steam in the boiler, the diameter and length of the pipe, the size and position of the throttle (if any), and the point of cut-off.

On a locomotive, the diameter of the driving-wheels, and the size of the blast orifice, the weight of the train, and the gradient, or curve.

On a condensing engine, the vacuum by the gauge, the kind

of condenser employed, the quantity of water used for one stroke of the engine, its temperature, and that of the discharge, the size of the air-pump and length of its stroke, whether single or double-acting, and, if driven independently of the engine, the number of its strokes per minute, and the height of the barometer.

The description of boiler used, the temperature of the feed-water, the consumption of fuel and of water per hour, and whether the boilers, pipes, and engine are protected from loss of heat by radiation; and if so, to what extent.

In addition to these, there are often special circumstances which should be noted.

IV. How to Keep the Indicator in Order.

The Indicator will not continue to work well unless it is kept in good order. When used, it generally becomes full of water, which will rust and thus weaken the spring; and the steam often contains impurities and grit, a portion of which is logded in it. After the Indicator has been used, and before putting up, unscrew the cover of the cylinder-case and draw off the upper ferule, with the pencil movement and the piston and spring attached, empty the water from the cylinder-case, carefully clean and dry all the parts, and replace them, lubricating the cylinder with a few drops of oil which is entirely free from gum. The cylinder is not to be removed from the case under any circumstances; the operation above directed gives complete access to it.

Sometimes the surfaces of the piston and cylinder become scratched or roughened by impurities in the steam, a fact which will be detected at once in the diagram by the unsteadiness of the line. If this shows the existence of any obstruction to the perfectly free action of the Indicator, take the instrument apart, as for cleaning, and remove the spring, then replace the piston in the cylinder, after cleaning and lubricating them; screw on the cover to guide the stem, and rub the piston up and down in the cylinder, at the same time revolving the stem between the thumb and finger. The surfaces will quickly wear each other smooth; no grinding or polishing material should be used, but the piston should be taken out once or twice during the operation, and the surfaces cleaned. The piston, if dry, ought to drop perfectly free from every position. Before re-

placing, lift the levers and let them fall, to see if their action also is entirely free. Then replace everything, taking care to screw the heads of the spring firmly up to the piston and cover. The paper cylinder requires to be lubricated occasionally with a drop or two of pure oil, applied at the end of the arbor, also the leading pulleys and the joints of the pencil movement.

V. How to Ascertain if the Action is free from Friction.

If the Indicator is impeded by friction it will not tell the truth. It is important, therefore, that its action should be frictionless, and also that the operator should know that it is so. He can learn its state in this respect by the following method:—Set the Indicator on the engine, fix a sheet of paper, and see that the marking-point is fine. Press the pencil down firmly with the hand, and let it return very slowly to a state of rest, and draw the atmospheric line. Then press it upward, and let it return in the same manner, and draw this line again. If proper care has been taken, and the pencil retraces the same fine line, the action is free from friction. Of course the pencil must not be vibrated, or suddenly released, but the pressure of the hand must be withdrawn in the most gradual manner. If impediments exist to its free motion, the double effect is shown of their resistance in both directions. Sometimes the lines come a quarter of an inch or more apart. The instrument is not in proper order for use if it will not perfectly stand this test. If the action is frictionless at the atmospheric line it is probably so everywhere, but for the upper part of the scale this may be tested by removing the spring, and after connecting the piston-rod and levers, observing if the pencil will drop *absolutely* free from any point at which it is released.

VI. How to Change the Springs.

Unscrew the coupling from the end of the piston-stem, the cover from the cylinder-case, and the spring from the piston and cover, introduce the new spring, and screw all up firmly again.

The lengths of the springs for the different scales are so proportioned to each other, that the pencil will always come to the proper position for drawing the atmospheric line. In putting

in the spring No. 1, the head from which the barrel projects to stop the compression of the spring should be screwed to the cover and not to the piston. Be careful that the heads are screwed up firmly to the piston and cover.

The spring which gives reaction to the paper cylinder is liable to break after considerable use, especially on engines running at high speeds, for which reason this cylinder should never be left to run unnecessarily. When breakage occurs a new spring can be readily substituted as follows:—Set the Indicator on the engine, if there are no other convenient means for holding it firmly, and remove the cover of the spring-case and the broken spring. Then hook the new spring on to the hook projecting from the ferule on the arbor, coil it into the case, and hook the end on the rim; see that it is coiled in the same direction with the cord. If the spring has not sufficient strength to keep the cord quite tight, another coil must be given to it, but it should not be coiled any tighter than is necessary for this purpose.

When the Indicator is applied to pumps, or when it is exposed only to the temperature of the atmosphere, an addition must be made to the pressure shown of one pound in each 40 lbs. That is, a spring when hot will indicate a pressure of 40 lbs., and when cold will show only 39 lbs.; and 40 lbs. is the real pressure in both cases. (*See* page 31.)

Prices of Springs and Metallic Paper.

The springs and metallic paper may be procured from any parties who sell these Indicators. The springs will be sent free by post to any address, on receipt of the price in stamps.

Prices.	£	s.	d.
Piston springs, with boxwood scales, each . .	0	10	0
Paper cylinder springs	0	1	6
Quire of metallic paper, cut into 360 diagram sheets	0	4	0

Indicators for special purposes, or containing special modifications, will be made on application.

PART SECOND.

TABLES.

INTRODUCTORY REMARKS.

We are accustomed to express the elastic force of steam in three ways—namely, in pounds of pressure that it exerts on the square inch, in the height of the column of mercury which it sustains, and in atmospheres. The actual pressure of the atmosphere is continually varying, the barometric column fluctuating generally between 28·5 and 30·5 inches in height; these points in either direction being, however, but rarely reached, and still more rarely passed. For scientific purposes, however, it is necessary that an exact pressure should be fixed upon as constituting an atmosphere, and the scientific world have agreed to employ for this purpose the French measure of a column of mercury 760 millimètres in height, at the temperature of 0° centigrade, or 32° Fahrenheit; which is indeed as nearly as possible the mean atmospheric pressure.

This pressure cannot be accurately expressed in English measurement, but only approximately by the use of decimals 760 millimètres being 29·9218004 inches of mercury, equal to a pressure on the square inch of 14·696303 lbs., very nearly.

It is common to say that an atmosphere is 15 lbs. on the square inch, or 30 inches of mercury, and in the Royal Arsenal of England at Woolwich the steam-engines, 13 in number, were a few years ago, when seen by the author, all provided with mercurial vacuum gauges, graduated in inches of mercury, and also in pounds on the square inch, each pound coinciding with an even number on the scale of inches.

This is shamefully rude, the pressure of 15 lbs. on the square inch being equal to that of a column of mercury 30·54 inches in height. It is sufficiently exact to say that 1 lb. on the square inch is equal to 2·036 inches of mercury, but it is a pity that we cannot employ in popular use a measure which, on

account of its simplicity and convenience, men of science have everywhere adopted.

This remark applies also, and with even greater force, to the thermometer. In the centigrade scale, the freezing-point—or the temperature at which, under the pressure of 760 millimètres of mercury at 0°, water passes from the fluid into the solid state—is taken as 0, and the boiling-point, or the temperature at which under the same pressure water passes most rapidly from the fluid into the gaseous state, or the highest temperature to which under this pressure it can be raised, is taken as 100, and the interval is divided into 100 equal parts, whence the name, signifying 100 steps. In contrast with this philosophical and simple measure is the Fahrenheit scale, which begins in a blunder and ends nowhere. As 32 degrees below the freezing-point was the lowest degree of cold that Fahrenheit himself was able to produce, he, in the true spirit of speculative reasoning, concluded that of course it was the lowest degree of cold that Almighty Power could produce, and assumed it as the absolute zero. We wonder how it was possible he could have been so ignorant of common actual temperatures; but doubtless we ourselves hold for facts some notions involving a degree of ignorance which will hereafter be wondered at quite as much. The freezing and boiling points, 32° and 212° of the Fahrenheit scale, originally inexact enough, have been made to coincide with the 0° and 100° of the centigrade scale, so that to reduce degrees of the latter to those of the former we have *only* to multiply them by 1·8 and add 32. If the Fahrenheit scale could be at once abolished, and the centigrade scale substituted in its place, we should be rid of an arbitrary nuisance, which is already being banished from the domain of science, and is a real impediment to popular scientific advancement.

In the following Tables it is attempted to present in English measures, and in a form entirely complete, the results of the experiments of M. Regnault upon the properties of steam. These, together with the results of the several series of preliminary experiments conducted for the purpose of ascertaining with exactness all the physical conditions necessary to be assumed in the course of the former, and descriptions of the apparatus and methods employed by him, have been before the world for upwards of twenty-five years, and have, it is believed, received the unanimous approval of competent judges. These experiments were conducted at the expense of the French Government

and under the auspices of the Academy of Sciences, and on account of the unprecedented care taken in ascertaining the conditions necessary to be assumed, the superior means and methods employed, and the wide and exhaustive range taken, there can be no doubt that the results obtained by M. Regnault are more nearly correct than any of those previously received, and it can hardly be imagined that any experiments which may hereafter be undertaken can command in anything like the same degree the confidence of men of science.

These experiments first established the now accepted fact, that the total heat of steam, or the sum of its sensible and latent heat, is not, as had before been supposed, the same for all temperatures, but increases with the increase of temperature, in the uniform ratio of ·305 of a degree of each degree of sensible heat; so that, as steam expands in volume, of each 1° of temperature that it loses, 695 parts only become latent, or are converted into internal work, and the remaining 305 parts are set free, and are capable of being converted into mechanical work.

The following data are given for the explanation and verifi cation of the Tables:—

Units.

All measurements, of whatever nature, are referred to some conventional unit. With some of these units our readers will need to be familiar, and they are here presented together. It is generally important to know the relation between the British and the French units, for which reason some of the latter also are given.

All units, except that of temperature, are now conventional and arbitrary. Attempts to employ natural constants as standards of reference have, in other departments, been wisely abandoned. The length of the pendulum vibrating seconds, for example, was a very admirable standard of measure in theory; but when the standard determined by that method had been burned up, it was found that it could not be restored by the same method, as the law directed it should be: first, because errors of unknown amount existed in the original determination; and, second because the degree of uncertainty involved in such an inquiry is incomparably greater than that involved in measurement. So, likewise, with respect to the new standard pound; the

Commission ascertained that fussing with water, as the law directed, would involve nearly 3000-fold the liability to error that mere weighing would do. So the new standard yard and pound are the closest possible restorations of the old ones, made by comparisons, conducted with infinite pains, with the best authenticated copies of them.

The attempt of the French Government, in the latter part of the last century, to make the $\frac{1}{10000000}$th part of the quadrant of the meridian circle the unit of the metric system, resulted in the establishment by law of a distance marked on a metal rod as the mètre, about which we know that it is *not* the $\frac{1}{10000000}$th part of the quadrant, but that it makes just as good a standard of measure as if it were.

So the kilogramme is, theoretically, the weight of a cubic décimètre of water,* but practically it is a block of platinum in the Archives at Paris, which, if it were to be lost, could not be precisely restored by weighing water till Doomsday.

The British standard of measure is the Imperial Standard Yard, kept in the Exchequer at Westminster.

The unit of linear measurement used in mechanics is the inch, or one thirty-sixth part of the Imperial standard yard.

The corresponding French standard is the mètre.

1 Mètre	=	39·37079	inches, approximately.
1 Décimètre	=	3·937079	„ „
1 Centimètre	=	0·3937079	„ „
1 Millimètre	=	0·03937079	„ „
1 cubic Centimètre	=	·0610270515	of a cubic inch,
	or	·0000353166	„ foot.
1 cubic Décimètre ..	=	1000 times the above, or	
		61·0270515 cubic inches,	
	or	·0353166 of a cubic foot.	

The unit of weight, pressure, or force, is the pound avoirdupois, consisting of 7000 grains, of which 5760 constitute a pound troy. The Imperial standard pound avoirdupois is a cylinder of platinum, kept in the Exchequer at Westminster.

* The term "water" will always be used here, unless otherwise expressed, to mean that fluid at its greatest density, which has been fixed by the experiments of Playfair and Joule, at 3·945° centigrade, or 39·101° Fahrenheit, and made perfectly pure, and free from air and gases, by distillation—theoretical water, which can only be approximated in reality, though without doubt pretty closely.

The French unit of weight is the gramme, which is in theory the weight of a cubic centimètre of water. The kilogramme, or 1 000 grammes, is the weight of a litre, or cubic décimètre, of water. The real standard is a piece of platinum, deposited in the Archives at Paris in 1799, and known as the "Kilogramme des Archives." The Commission appointed to restore the lost British standards went to Paris and weighed this piece of platinum, and found out by that means the weight of a cubic foot of water.

This is the way in which this was arrived at:—They found the kilogramme to weigh 2·20462125 of the standard platinum pounds. It seems as if it would be a very simple matter to determine what a cubic foot of water weighs, the method being to weigh, first in the air and then in the water, a solid of absolutely known dimensions, when the weight that it loses by the immersion and that which it at first lost by immersion in the air being added together, their sum is the weight of the water that it displaces; but the Commissions appointed by five European Governments to ascertain the weight of water arrived each at a different result.

The following are the weights of an English cubic foot of water, at 62° Fahrenheit, as determined by the different Commissions respectively:—

French	62·3566 lbs.
English	62·3860 „
Swedish	62·3746 „
Austrian	62·3347 „
Russian	62·3556 „

To obtain these weights at the greatest density of water, the above must be multiplied by 1·00109. Now it is really of little consequence what the weight of a given bulk of water is to the last decimal; but it is of great consequence that the scientific world should agree on what it is. So, without discussing about the very small and uncertain amount of error involved in the French determination, it is assumed to be correct, and then a simple division gives the weight of a cubic foot of water; for if ·0353166 of a cubic foot weighs 2·20462125 lbs. then 1 cubic foot weighs 62·4245 lbs.: not precisely, but nearly enough for all practical purposes. This is a determination of great importance. By means of it, for example, the elastic forces of steam at different temperatures are reduced from

millimètres of mercury, in which they are given by Regnault, to pounds on the square inch.

The unit of work or power * is 1 lb. lifted 12 inches, or 1 lb. of force acting through 1 foot of distance, and is called the foot-pound.

33,000 foot-pounds, or units of work, performed in one minute, make a horse-power.

The French unit of work is 1 kilogramme lifted 1 mètre, called the kilogramme-mètre, or, for brevity, kilo-mètre. It is equal to 7·233136 foot-pounds. 75 kilogramme-mètres exerted in one second constitute the French horse-power, equal to 32,549·112 foot-pounds per minute. The English horse-power is therefore 1·01416 French horse-powers, and the French horse-power is ·986 of an English horse-power.

The unit of elasticity, by which the expansive force exerted by elastic fluids is measured, is, for popular use, 1 lb. on 1 square inch. The scientific unit of elasticity is 1 atmosphere.

1 atmosphere is equal to	760·	millimètres	of mercury
	or to 29·9218004	inches	„
	or to 406·814704	„	water
	or to 14·696303	lbs. on the sq. inch.	
1 lb. on the sq. in. is equal to	27·68143	inches	of water
	or to 2·03601	„	mercury
	or to 51·7137	millimètres	„

The unit of temperature is the degree Fahrenheit, or $\frac{1}{180}$th part of the distance on the thermometric scale between the freezing and the boiling points of water, under the pressure of one atmosphere. The corresponding French unit is the degree centigrade, or $\frac{1}{100}$th part of the same distance.

The unit of heat, or the thermal unit, is the quantity of heat necessary to be added to 1 lb. of water, at or near to its freezing-point, to raise its temperature 1° Fahrenheit. †

* The distinction should be sharply apprehended between force and power, the former being a static conception, and the latter a dynamic expression of the union of force with motion.

† The thermal unit is given by Rankine as the quantity of heat which corresponds to an interval of 1° in the temperature of 1 lb. of water *at its greatest density* (39·1°). The difference between this statement and that given in the text is small, though not insensible; but the latter seems the correct one. Water at 0° centigrade, or 32° Fahrenheit, is the unit of comparison employed for all measurements of the capacities for heat of all sub-

The quantity of heat required to raise the temperature of water 1° increases with the increase of its temperature, and that in an increasing ratio. This increase has been determined with precision by M. Regnault, and is here shown in Table No. II. (of the specific heat of water), and in the revised tables of the properties of steam. The figures expressing these quantities have been carried to the third place of decimals, and will meet all requirements of exactness.

One unit of heat is equivalent to 772 units of work. This is known as the mechanical equivalent of heat, or, from the physicist by whose investigations this relation has been established, "Joule's Equivalent."

The French unit of heat is the quantity required to raise the temperature of 1 kilogramme of water, at or near to its freezing-point, 1° centigrade. One French is equal to 3·96832 British units, and one British is ·251996 of a French unit of heat.

The specific heat of a body is the quantity of heat necessary to be imparted to it in order to raise its temperature 1°, relatively to that quantity that is required to raise by 1° the temperature of an equal weight of water at or about the temperature of 32°. The specific heat of water is greater than that of any other substance; so that, this being taken as 1·, that of any other substance is expressed in decimals.

The specific heat of superheated steam was investigated by M. Regnault, who ascertained it to be ·48051.

Saturated steam is steam, the density, the elastic force, and the temperature of which are unchangeable relatively to each other. If one of these properties is maintained constant, both of the others must be so also: and if one is changed, each of the others also changes in a fixed ratio. It is always at its dew-point; and if its density is maintained, all loss of heat which it suffers must be supplied by partial condensation. It does not

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stances whatever. If the specific heat of water were constant, then the thermal unit would be merely the quantity of heat required to raise the temperature of 1lb. of water 1°, which would be the same at whatever part of the scale; but since it is not constant, the unit must be the quantity so required at that temperature at which the specific heat of water is 1, and that is 32°. (See Regnault's Memoir "On the Specific Heat of Liquid Water at different Temperatures," the tenth Memoir in vol. xxi. of *Mémoires* of the French Academy.) The subject is one with which the density of water has no concern. It is immaterial in this regard what the volume of a pound of water may be.

necessarily contain any water, but its temperature can be no higher than that of the water from which it has been evaporated. Until the experiments of M. Regnault, its specific heat was supposed to be 0; that is, it was supposed that no additional heat was necessary to be imparted to it in order to raise its temperature, but that the increase of temperature was wholly supplied from the latent heat, which became reduced in the same degree: the total heat which it contained, sensible and latent, being thus the same for all temperatures. But the experiments of M. Regnault have shown that an addition to its total heat of ·305 of a degree is necessary in order to raise its temperature, or sensible heat, 1°. The specific heat of saturated steam is therefore ·305.

By total heat is not meant, of course, the total heat reckoned from the absolute zero, which has been fixed at 461·2° below the zero of Fahrenheit, but the total heat counting from the latter point.

The unit of angular, or trigonometrical measurement, is the radius of the circle to which the sides of the triangle are referred.

The unit of specific gravity is the weight of water. The specific gravity of a body is its weight, at the temperature of 0° centigrade, or 32° Fahrenheit, compared with that of an equal volume of water.

For measuring the specific gravity, or comparative density, of steam and gases, the atmosphere, at the temperature of 32° Fahrenheit, and sustaining a column of 760 millimètres of mercury, is employed as the unit.

The specific gravity of water being . . .	1·
That of air, under 760 mm. of mercury, is .	·001293187
And that of mercury	13·59593

The volume of water being	1·
That of the same weight of air at 32° is . .	773·283
And that of the same weight of mercury at 32°	·0735514

The volume of 1 lb. of water is 27·68143 cubic inches, or ·01602 of a cubic foot.

The weight of a cubic foot of water is . .	62.4245 lbs.
„ „ „ „ air	.0807265 lbs.
„ „ „ inch of water . . .	.0361253 lbs.
„ „ „ „ mercury . .	·491157 lbs.

EXPANSION.

1. *The expansion of water.*—The rate of expansion of water by heat varies more than that of any other substance. Between

TABLE I.

Showing the expansion of Water by Heat, between its point of maximum density and its boiling point.

Centigrade.	Volumes as given by Kopp.	Corrected Volumes.	1st Difference.	2nd Difference.
4°	1·00000	1·00000		
5	1·00001	1·00001		
10	1·00025	1·00025	24	34
15	1·00082	1·00083	58	30
20	1·00169	1·00171	88	27
25	1·00284	1·00286	115	24
30	1·00423	1·00425	139	22
35	1·00583	1·00586	161	20
40	1.00768	1·00767	181	19
45	1·00967	1·00967	200	19
50	1·01190	1·01186	219	18
55	1·01423	1·01423	237	18
60	1·01672	1·01678	255	18
65	1·01943	1·01951	273	17
70	1·02238	1·02241	290	17
75	1·02554	1·02548	307	17
80	1·02871	1·02872	324	17
85	1·03202	1·03213	341	16
90	1·03553	1·03570	357	16
95	1·03921	1·03943	373	16
100°	1·04312	1·04332	389	

39·1° and 212° its volume increases from 1· to 1·04332; and its expansion, for each 1° added to its temperature, increases from

0 at 40° to 00044 at 212°. Above the latter point nothing is known about it.

The author has ventured to make the slight changes in the best tables of the volume of water at different temperatures, which are necessary to cause the second differences to fall into the following series of ordinates of a curve, which must be very nearly correct, and the prolongation of which should give the volumes at higher temperatures. The volumes are given at intervals of 5° centigrade.

2. *The expansion of mercury.*—The expansion of mercury for each increase of 1° in its temperature, the volume at 32° being 1·, is ·00010085.

The pressure of the atmosphere, and the elastic force of steam and gases, are measured by the height of the column of mercury at 32° which they sustain.

In making such observations, therefore, the temperature of the mercury must be taken, and the correction made for its expansion, as above given.

This requirement is sometimes forgotten, as notably by Mr. Isherwood, in some of his experiments, which were a few years ago conducted at great cost to the American Government, and for a time attracted some attention, but are now themselves forgotten.

The refined investigations of M. Regnault showed the coefficient of the expansion of mercury to increase slightly with the increase in its temperature, but the above is sufficiently exact for all practical requirements.

3. *The expansion of elastic fluids.*—This is produced in two ways: 1st, by the removal of pressure from them while their temperature remains unchanged; and 2nd, by the elevation of their temperature.

The law of the gases, known as Boyle's law, or the law of Mariotte, is stated by Regnault as follows:—"The volumes of a given weight of a gas, at a constant temperature, are inversely proportional to the pressures which the gas sustains; or, in other terms, the densities of the gas, at the same temperature, are proportional to the pressure." The theory of the law is, that gas being perfectly elastic, its density must vary directly, and its volume inversely, as the pressures to which it is subjected. "We are accustomed," says Regnault, "to regard the

law of Mariotte as the mechanical expression of the perfectly gaseous state."

This celebrated law was first announced independently in the latter part of the 17th century, by the English philosopher Boyle, and by the French Abbé Mariotte. Up to the time of Regnault, although other gases had been found to show a degree of compressibility more or less greater than according to this law they ought to do, and so to be more or less imperfectly elastic, the conviction had become general, as the result of experiments made by the most eminent investigators, that atmospheric air followed this law rigorously.

Regnault, sharing this belief, first alludes to the matter incidentally in discussing the results of a series of experiments conducted for the purpose of determining the density of gases. He observes, with respect to air, that "the calculation gives constantly a density a little greater than the experiment, but the differences are too small to be attributed to anything except errors of observation." Further consideration of these and other results, however, led him to feel such a degree of uncertainty on the subject, that, "notwithstanding the imposing authority of the physicists by whose experiments the absolute conformity of air to the law of Mariotte seemed to be demonstrated in an incontestable manner," he determined to enter upon new researches.

His experiments took a wide range, and will long be studied as models of the method by which certainty is to be reached. It pertains to our subject only to state here very briefly their results respecting air, which of all gases showed by far the closest approach to perfect elasticity.

These are given in the following Table:—

Volume.	Elastic Force.	Deficiency as compared with the force calculated by the law of Mariotte.	Percentage of the calculated force which the deficiency amounts to.
1	1·0000		
$\frac{1}{5}$	4·9794	·0206	·412
$\frac{1}{10}$	9·9162	·0838	·838
$\frac{1}{15}$	14·8248	·1752	1·168
$\frac{1}{20}$	19·7198	·2802	1·401

This deviation is so small that, although in a theoretical point of view it is highly important to be known, it may in practice be disregarded, and air be considered as a perfect gas.

The expansion of gases by heat was also re-investigated by M. Regnault. Two modes of experimenting were followed by him: one, to ascertain the change of elastic force which a given volume of gas undergoes in passing from the freezing to the boiling point of water, from which the change of volume was to be calculated, which, according to the law of Mariotte, would be equivalent to this change of elastic force; and the other, to ascertain directly the change of volume which a given weight of gas undergoes in passing through the same change of temperature, the elastic force remaining constant. If gases followed the law of Mariotte precisely, the result by either of these methods should be the same; but since they do not, the latter method only gives the actual expansion. The expansion of air caused by raising its temperature from 32° to 212° was found by him to be,—

When calculated from the increased pressure exerted under constant volume, ·3665.	When the enlargement of volume is observed directly under constant pressure, ·3670.

From the latter of these we find the co-efficient of the expansion of dry air for each increase of 1° Fahrenheit to be ·002039, and that it doubles its volume for each increase in its temperature of 272·48° centigrade, or 490·4° Fahrenheit.

In the case of saturated steam this difficulty is found to exist, that the expansion and contraction of its volume consequent on changes of pressure and those consequent on changes of temperature cannot be investigated separately. At the same time that its volume is being contracted by increasing pressure it is being expanded by increasing temperature; and so, on the other hand, while expanding as it is relieved from confining pressure, it is contracting, or, since it must fill a given capacity, is losing its expansive force in a corresponding degree, by reduction of its temperature.

For some unexplained reason this subject was not investigated by M. Regnault. It is certain that he originally intended to make it the subject of experiment. Indeed, it was a part of the task assigned him, which was "to determine the principal laws and numerical conditions which enter into the calculation of steam-engines."

In his general introduction he says:—"But in order to calculate, in each case, the weight of a cubic metre of vapour, it is necessary to know: 1. The laws according to which the density of saturated steam varies under different pressures; and, 2. The coefficient of the dilatation (or expansion by heat) of the vapour of water taken in different states of density. Mechanicians admit, for the most part, that the weight of a cubic metre of vapour of given pressure and temperature may be calculated by applying to it the law of Mariotte, and the law of the uniform dilatation of gases. But these laws are not rigorously exact even for the permanent gases, and it is to be feared that they may be completely false for saturated vapours."

Again, at the opening of his memoir upon the determination of the density of gases he observes:—"The necessity for determining with precision the density of the vapour of water, under different circumstances of temperature and pressure, has led me to study the methods which are employed to determine the densities of elastic fluids." But, after all, he seems never to have reached this branch of his vast subject.

Mr. Rankine has computed the density of steam from its latent heat of evaporation. He finds the densities at different temperatures to be greater than those corresponding to the perfectly gaseous state. "For steam at low temperatures," he remarks, "the difference is trifling, but increases rapidly as the pressure increases."

Mr. Rankine's computation of the weight of steam has been here followed for pressures below 16 lbs. For higher pressures, the weight and corresponding specific volume are taken from the Table of Fairbairn and Tate, which there agrees pretty closely with that of Mr. Rankine.

In the former editions of this treatise the weight and specific volume of steam have been given, as computed according to the gaseous laws. Subsequent observation seems, however, to show that its density increases more rapidly than these call for, and probably nearly in the ratio given by the above-named authorities. A degree of uncertainty still invests the subject. The single experimental determination of the specific volume of steam, at 212° and under one atmosphere of pressure, from which former tables have been computed, is itself rejected. Mr. Rankine says, "There is no direct experimental determination of the exact amount or law of the excess" (in the

weight of steam above that which the gaseous laws would seem to require). "The general law which it follows is unknown." There seems to be a real need of experiments on this subject which shall be conclusive in their nature, but the difficulties which surround it are such that these are scarcely to be hoped for.

It seems necessary to be stated that there is no such thing as steam gas, or gasification, or heat of gasification, in the sense in which these terms have been employed, as distinguished, on the one hand, from simple steam and evaporation, and heat of vaporisation; and, on the other hand, from superheated steam.

Steam exists only as saturated and as superheated steam. The number of thermal units contained in the former is given in the following Tables. The additional number contained in the latter is found by multiplying the degrees of superheat—by which the temperature exceeds that of saturated steam under the same pressure—by the decimal ·48051. Experiments have proved that all the heat abandoned by steam, when condensed, is thus accounted for.

TABLE II.

Giving the Number of Thermal Units contained in One Pound of Water, at temperatures from 35° to 390° Fahrenheit.

Temperature.	Number of Thermal Units.	Increase.	Temperature.	Number of Thermal Units.	Increase.	Temperature.	Number of Thermal Units.	Increase.
35°	35·000		155°	155·339	5·034	275°	276·985	5·107
40	40·001	5·001	160	160·374	5·035	280	282·095	5·110
45	45·002	5.001	165	165·413	5·039	285	287·210	5·115
50	50·003	5·001	170	170·453	5·040	290	292·329	5·119
55	55·006	5·003	175	175·497	5·044	295	297·452	5·123
60	60·009	5·003	180	180·542	5·045	300	302·580	5·128
65	65·014	5·005	185	185·591	5·049	305	307·712	5·132
70	70·020	5·006	190	190·643	5·052	310	312·848	5·136
75	75·027	5·007	195	195·697	5·054	315	317·988	5·140
80	80·036	5·009	200	200·753	5·056	320	323·134	5·146
85	85·045	5·009	205	205·813	5·060	325	328·284	5·150
90	90·055	5·010	210	210·874	5·061	330	333·438	5·154
95	95·067	5·012	215	215·939	5·065	335	338·596	5·158
100	100·080	5·013	220	221·007	5·068	340	343·759	5·163
105	105·095	5·015	225	226·078	5·071	345	348·927	5·168
110	110·110	5·015	230	231·153	5·075	350	354·101	5·174
115	115·129	5·019	235	236·232	5·079	355	359·280	5·179
120	120·149	5·020	240	241·313	5·081	360	364·464	5·184
125	125·169	5·020	245	246·398	5·085	365	369·653	5·189
130	130·192	5·023	250	251·487	5·089	370	374·846	5·193
135	135·217	5·025	255	256·579	5·092	375	380·044	5·198
140	140·245	5·028	260	261·674	5·095	380	385·247	5·203
145	145·275	5·030	265	266·774	5·100	385	390·456	5·209
150°	150·305	5·030	270°	271·878	5·104	390°	395·672	5·216

TABLE III.

THE PROPERTIES OF SATURATED STEAM AT EACH DEGREE OF TEMPERATURE, FROM 32° TO 213·3° FAHRENHEIT, OR FROM ·089 OF A POUND TO 15 POUNDS' PRESSURE ON THE SQUARE INCH.

Temperature of the vapour, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.				Elastic Force; expressed in:			Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	Number contained in water.	Number required for evaporation; known as latent heat, or heat of vaporisation.	Total number contained in the vapour.	Difference.	Pounds on the square inch.	Inches of mercury at 32°.	Millimètres of mercury at 32°.	Volume.	Decrease of Volume.		
32°	32·000	1091·700	1123·700	·305	·089	·1811	4·6	208080	7600	·00030	·0037
33	33·000	1091·005	1124·005	·305	·092	·1884	4·786	200480	7300	·00030	·0039
34	34·000	1090·310	1124·310	·305	·096	·1960	4·979	193180	7000	·00031	·0040
35	35·000	1089·615	1124·615	·305	·100	·2039	5·179	186180	6800	·00032	·0042
36	36·000	1088·920	1124·920	·305	·104	·2121	5·386	179380	6600	·00033	·0044
37	37·000	1088·225	1125·225	·305	·108	·2205	5·6	172780	6400	·00034	·0045
38	38·000	1087·530	1125·530	·305	·112	·2292	5·822	166380	6150	·00036	·0047
39	39·001	1086·834	1125·835	·305	·117	·2382	6·051	160230	5900	·00038	·0049
40	40·001	1086·139	1126·140	·305	·122	·2476	6·288	154330	5710	·00040	·0051
41	41·001	1085·444	1126·445	·305	.127	·2573	6·534	148620	5400	·00042	·0052
42	42·001	1084·749	1126·750	·305	·132	·2673	6·789	143220	5150	·00043	·0054
43	43·001	1084·054	1127·055	·305	·137	·2777	7·053	138070	4950	·00045	·0056
44	44·002	1083·358	1127·360	·305	·142	·2884	7·325	133120	4750	·00047	·0058
45	45·002	1082·663	1127·665	·305	·147	·2994	7·606	128370	4530	·00049	·0060
46	46·002	1081·968	1127·970	·305	·152	·3109	7·897	123840	4230	·00050	·0063
47	47·002	1081·273	1128·275	·305	·158	·3228	8·199	119610	4120	·00052	·0065
48	48·003	1080·577	1128·580	·305	·164	·3351	8·511	115490	4020	·00054	·0068

49	49·003	1079·882	1128·885	·305	·170	·3478	8·833	111470	3840	·00056	·0070
50	50·003	1079·187	1129·190	·305	·176	·3608	9·165	107630	3700	·00058	·0072
51	51·004	1078·491	1129·495	·305	·183	·3743	9·508	103930	3600	·00060	·0074
52	52·004	1077·796	1129·800	·305	·190	·3883	9·864	100330	3400	·00062	·0077
53	53·005	1077·100	1130·105	·305	·197	·4028	10·232	96930	3250	·00065	·0079
54	54·005	1076·405	1130·410	·305	·205	·4177	10·611	93680	3140	·00067	·0082
55	55·006	1075·709	1130·715	·305	·212	·4332	11·002	90540	3040	·00069	·0085
56	56·006	1075·014	1131·020	·305	·220	·4492	11·406	87500	2940	·00071	·0088
57	57·007	1074·318	1131·325	·305	·228	·4656	11·823	84560	2820	·00073	·0091
58	58·007	1073·623	1131·630	·305	·236	·4825	12·254	81740	2720	·00076	·0094
59	59·008	1072·927	1131·935	·305	·245	·5000	12·699	79020	2650	·00079	·0098
60	60·009	1072·231	1132·240	·305	·254	·5180	13·159	76370	2560	·00082	·0101
61	61·010	1071·535	1132·545	·305	·263	·5367	13·633	73810	2480	·00085	·0105
62	62·011	1070·839	1132·850	·305	·273	·5560	14·122	71330	2390	·00088	·0109
63	63·012	1070·143	1133·155	·305	·282	·5758	14·626	68940	2310	·00091	·0113
64	64·013	1069·447	1133·460	·305	·292	·5962	15·146	66630	2210	·00094	·0117
65	65·014	1068·751	1133·765	·305	·302	·6173	15·682	64420	2130	·00097	·0121
66	66·015	1068·055	1134·070	·305	·313	·6391	16·234	62290	2010	·00100	·0125
67	67·016	1067·359	1134·375	·305	·324	·6615	16·803	60280	1940	·00103	·0130
68	68·018	1066·662	1134·680	·305	·335	·6846	17·390	58340	1870	·00107	·0134
69	69·019	1065·966	1134·985	·305	·347	·7084	17·996	56470	1810	·00111	·0138
70	70·020	1065·270	1135·290	·305	·359	·7330	18·621	54660	1750	·00115	·0142
71	71·021	1064·574	1135·595	·305	·372	·7583	19·265	52910	1700	·00119	·0147
72	72·023	1063·877	1135·900	·305	·385	·7844	19·928	51210	1640	·00123	·0152
73	73·024	1063·181	1136·205	·305	·398	·8114	20·611	49570	1570	·00127	·0157
74	74·026	1062·484	1136·510	·305	·411	·8391	21·314	48000	1490	·00131	·0162
75°	75·027	1061·788	1136·815	·305	·425	·8676	22·037	46510	1450	·00135	·0167

TABLE III.—*continued.*

Temperature of the vapour, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.				Elastic Force; expressed in:			Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	Number contained in the water.	Number required for evaporation; known as latent heat, or heat of vaporisation.	Total number contained in the vapour.	Difference.	Pounds on the square inch.	Inches of mercury at 32°.	Millimètres of mercury at 32°.	Volume.	Decrease of Volume.		
76°	76·029	1061·091	1137·120	·305	·440	·8969	22·782	45060	1410	·00139	·0172
77	77·030	1060·395	1137·425	·305	·455	·9271	23·550	43650	1370	·00143	·0177
78	78·032	1059·698	1137·730	·305	·470	·9583	24·341	42280	1320	·00148	·0183
79	79·034	1059·001	1138·035	·305	·486	·9905	25·155	40960	1270	·00153	·0189
80	80·036	1058·304	1138·340	·305	·502	1·023	25·993	39690	1210	·00158	·0195
81	81·037	1057·608	1138·645	·305	·518	1·056	26·855	38480	1160	·00163	·0201
82	82·039	1056·911	1138·950	·305	·535	1·091	27·741	37320	1130	·00168	·0208
83	83.041	1056·214	1139·255	·305	·553	1·127	28·653	36190	1090	·00173	·0214
84	84·043	1055·517	1139·560	·305	·571	1·163	29·591	35100	1050	·00178	·0220
85	85·045	1054·820	1139·865	·305	·590	1·201	30·556	34050	1020	·00183	·0227
86	86·047	1054·123	1140·170	·305	·609	1·240	31·548	33030	980	·00189	·0234
87	87·049	1053·426	1140·475	·305	·629	1·281	32·568	32050	950	·00195	·0241
88	88·051	1052·729	1140·780	·305	·650	1·323	33·616	31100	920	·00201	·0249
89	89·053	1052·032	1141·085	·305	·671	1·366	34·694	30180	890	·00207	·0256
90	90·055	1051·335	1141·390	·305	·692	1·410	35·802	29290	860	·00213	·0264
91	91·057	1050·638	1141·695	·305	·715	1·454	36·942	28430	830	·00219	·0272
92	92·059	1049·941	1142·000	·305	·738	1·500	38·113	27600	800	·00226	·0280
93	93·061	1049·244	1142·305	·305	·761	1·548	39·317	26800	780	·00233	·0288
94	94·063	1048·547	1142·610	·305	·785	1·597	40·555	26020	750	·00240	·0297
95	95·065	1047·850	1142·915	·305	·809	1·647	41·827	25270	730	·00247	·0306

96	96·068	1047·152	1143·220	·305	·834	1·698	43·134	24540	710	·00254	·0315
97	97·071	1046·454	1143·525	·305	·860	1·751	44·476	23830	690	·00262	·0324
98	98·074	1045·756	1143·830	·305	·887	1·805	45·854	23140	670	·00270	·0334
99	99·077	1045·058	1144·135	·305	·914	1·861	47·268	22470	640	·00278	·0344
100	100·080	1044·360	1144·440	·305	·943	1·918	48·719	21830	620	·00286	·0354
101	101·083	1043·662	1144 745	·305	·972	1·977	50·207	21210	590	·00294	·0364
102	102·086	1042·964	1145·050	·305	1·001	2·037	51·734	20620	570	·00302	·0374
103	103·089	1042·266	1145·355	·305	1·031	2·099	53·3	20050	550	·00311	·0385
104	104·092	1041·568	1145·660	·305	1·062	2·163	54·906	19500	530	·00320	·0396
105	105·095	1040·870	1145·965	·305	1·094	2·227	56·554	18970	510	·00330	·0408
106	106·098	1040·172	1146·270	·305	1·126	2·293	58·245	18460	500	·00340	·0421
107	107·101	1039·474	1146·575	·305	1·159	2·361	59·981	17960	490	·00350	·0434
108	108·104	1038·776	1146·880	·305	1·193	2·431	61·763	17470	480	·00360	·0446
109	109·107	1038·078	1147·185	·305	1·229	2·503	63·592	16990	470	·00370	·0458
110	110·110	1037·380	1147·490	·305	1·265	2·577	65·469	16520	450	·00380	·0470
111	111·113	1036·682	1147·795	·305	1·302	2·653	67·394	16070	430	·00390	·0483
112	112·117	1035·983	1148·100	·305	1·341	2·731	69·368	15640	420	·00400	·0496
113	113·121	1035·284	1148·405	·305	1·381	2·810	71·391	15220	400	·00410	·0509
114	114·125	1034·585	1148·710	·305	1·421	2·892	73·464	14820	390	·00421	·0522
115	115·129	1033·886	1149·015	·305	1·462	2·976	75·588	14430	380	·00433	·0535
116	116·133	1033·187	1149·320	·305	1·504	3·061	77·764	14050	370	·00445	·0551
117	117·137	1032·488	1149·625	·305	1·547	3·149	79·993	13680	360	·00457	·0567
118	118·141	1031·789	1149·930	·305	1·591	3·239	82·276	13320	350	·00470	·0583
119	119·145	1031·090	1150·235	·305	1·636	3·331	84·615	12970	340	·00483	·0599
120	120·149	1030·391	1150·540	·305	1·682	3·425	87·012	12630	330	·00496	·0615
121	121·153	1029·692	1150·845	·305	1·730	3·522	89·467	12300	320	·00508	·0631
122°	122·157	1028·993	1151·150	·305	1·779	3·621	91·982	11980	310	·00521	·0648

TABLE III.—*continued.*

Temperature of the vapour, and of the water from which is is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.				Elastic Force; expressed in:			Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	Number contained in the water.	Number required for evaporation; known as latent heat, or heat of vaporisation.	Total number contained in the vapour.	Difference.	Pounds on the square inch.	Inches of mercury at 32°.	Millimètres of mercury at 32°.	Volume.	Decrease of Volume.		
123°	123·161	1028·294	1151·455	·305	1·828	3·723	94·557	11670	300	·00535	·0665
124	124·165	1027·595	1151·760	·305	1·879	3·826	97·194	11370	290	·00549	·0682
125	125·169	1026·896	1152·065	·305	1·931	3·933	99·893	11080	280	·00563	·0699
126	126·173	1026·197	1152·370	·305	1·984	4·042	102·656	10800	270	·00578	·0716
127	127·177	1025·498	1152·675	·305	2·039	4·153	105·484	10530	265	·00593	·0734
128	128·182	1024·798	1152·980	·305	2·096	4·267	108·379	10265	255	·00608	·0753
129	129·187	1024·098	1153·285	·305	2·154	4·384	111·342	10010	250	·00624	·0773
130	130·192	1023·398	1153·590	·305	2·213	4·503	114·375	9760	244	·00640	·0793
131	131·197	1022·698	1153·895	·305	2·273	4·625	117·478	9516	240	·00656	·0813
132	132·202	1021·998	1154·200	·305	2·335	4·750	120·653	9276	230	·00673	·0834
133	133·207	1021·298	1154·505	·305	2·398	4·878	123·901	9046	220	·00690	·0855
134	134·212	1020·598	1154·810	·305	2·461	5·009	127·223	8826	215	·00707	·0876
135	135·217	1019·898	1155·115	·305	2·526	5·143	130·621	8611	210	·00725	·0898
136	136·222	1019·198	1155·420	·305	2·594	5·280	134·096	8401	210	·00743	·0921
137	137·227	1018·498	1155·725	·305	2·663	5·420	137 650	8191	200	·00761	·0944
138	138·233	1017·797	1156·030	·305	2·732	5·563	141·283	7991	193	·00780	·0968
139	139·239	1017·096	1156·335	·305	2·803	5·709	144·996	7798	185	·00799	·0992
140	140·245	1016·395	1156·640	·305	2·876	5·858	148·791	7613	180	·00819	·1016
141	141·251	1015·694	1156·945	·305	2·952	6·011	152·670	7433	175	·00839	·1041
142	142·257	1014·993	1157·250	·305	3·029	6·167	156·635	7258	170	·00860	·1066

143	143·263	1014·292	1157·555	·305	3·108	6·327	160·688	7088	168	·00881	·1092
144	144·269	1013·591	1157·860	·305	3·188	6·490	164·831	6920	165	·00903	·1119
145	145·275	1012·890	1158·165	·305	3·270	6·657	169·065	6755	160	·00925	·1147
146	146·281	1012·189	1158·470	·305	3·353	6·827	173·392	6595	155	·00948	·1175
147	147·287	1011·488	1158·775	·305	3·438	7·001	177·814	6440	150	·00971	·1203
148	148·293	1010·787	1159·080	·305	3·526	7·179	182·331	6290	146	·00993	·1232
149	149·299	1010·086	1159·385	·305	3·615	7·361	186·945	6144	140	·01016	·1261
150	150·305	1009·385	1159·690	·305	3·707	7·547	191·658	6004	137	·01040	·1290
151	151·311	1008·684	1159·995	·305	3·801	7·736	196·471	5867	133	·01064	·1320
152	152·318	1007·982	1160·300	·305	3·896	7·929	201·386	5734	130	·01089	·1350
153	153·325	1007·280	1160·605	·305	3·992	8·127	206·404	5604	127	·01114	·1381
154	154·332	1006·578	1160·910	·305	4·090	8·329	211·527	5477	124	·01140	·1413
155	155·339	1005·876	1161·215	·305	4·191	8·535	216·756	5353	121	·01167	·1446
156	156·346	1005·174	1161·520	·305	4·295	8·745	222·092	5232	118	·01194	·1480
157	157·353	1004·472	1161·825	·305	4·400	8·959	227·537	5114	114	·01222	·1514
158	158·360	1003·770	1162·130	·305	4·507	9·178	233·093	5000	112	·01250	·1550
159	159·367	1003·068	1162·435	·305	4·617	9·401	238·763	4888	109	·01279	·1587
160	160·374	1002·366	1162·740	·305	4·729	9·629	244·551	4779	106	·01308	·1624
161	161·381	1001·664	1163·045	·305	4 843	9·861	250·459	4673	104	·01338	·1662
162	162·389	1000·961	1163·350	·305	4·960	10·098	256·488	4569	102	·01368	·1700
163	163·397	1000·258	1163·655	·305	5·079	10·340	262·641	4467	99	·01399	·1738
164	164·405	999·555	1163·960	·305	5·200	10·588	268·919	4368	97	·01430	·1776
165	165·413	998·852	1164·265	·305	5·324	10·840	275·323	4271	94	·01462	·1815
166	166·421	998·149	1164·570	·305	5·451	11·097	281·855	4177	92	·01495	·1854
167	167·429	997·446	1164·875	·305	5·580	11·359	288·517	4085	89	·01528	·1893
168	168·437	996·743	1165·180	·305	5·711	11·627	295·311	3996	86	·01562	·1935
169	169·445	996·040	1165·485	·305	5·845	11·900	302·238	3910	84	·01596	·1978
170°	170·453	995·337	1165·790	·305	5·981	12·178	309·300	3826	82	·01631	·2022

TABLE III.—*continued.*

Temperature of the vapour, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.				Elastic Force; expressed in:			Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	Number contained in the water	Number required for evaporation; known as latent heat, or heat of vaporisation.	Total number contained in the vapour.	Difference.	Pounds on the square inch.	Inches of mercury at 32°.	Millimètres of mercury at 32.°	Volume.	Decrease of Volume.		
171°	171·461	994·634	1166·095	·305	6·120	12·461	316·499	3744	80	·01667	·2067
172	172·470	993·930	1166·400	·305	6·262	12·750	323·838	3664	78	·01704	·2112
173	173·479	993·226	1166·705	·305	6·408	13·045	331·319	3586	76	·01741	·2158
174	174·488	992·522	1167·010	·305	6·555	13·345	338·945	3510	74	·01779	·2204
175	175·497	991·818	1167·315	·305	6·704	13·651	346·719	3436	71	·01817	·2251
176	176·506	991·114	1167·620	·305	6·857	13·963	354·643	3365	70	·01855	·2299
177	177·515	990·410	1167·925	·305	7·013	14·281	362·719	3295	69	·01894	·2347
178	178·524	989·706	1168·230	·305	7·172	14·605	370·949	3226	67	·01934	·2398
179	179·533	989·002	1168·535	·305	7·335	14·935	379·335	3159	66	·01975	·2450
180	180·542	988·298	1168·840	·305	7·500	15·271	387·879	3093	64	·02017	·2502
181	181·551	987·594	1169·145	·305	7·668	15·614	396·583	3029	63	·02060	·2554
182	182·561	986·889	1169·450	·305	7·841	15·963	405·449	2966	61	·02104	·2607
183	183·571	986·184	1169·755	·305	8·016	16·318	414·479	2905	59	·02148	·2662
184	184·581	985·479	1170·060	·305	8·194	16·680	423·676	2846	57	·02193	·2718
185	185·591	984·774	1170·365	·305	8·375	17·049	433·041	2789	56	·02238	·2774
186	186·601	984·069	1170·670	·305	8·558	17·425	442·578	2733	55	·02284	·2831
187	187·611	983·364	1170·975	·305	8·745	17·807	452·291	2678	54	·02331	·2888
188	188·621	982·659	1171·280	·305	8·936	18·196	462·183	2624	53	·02379	·2948
189	189·632	981·953	1171·585	·305	9·132	18·593	472·257	2571	52	·02428	·3009
190	190·643	981·247	1171·890	·305	9·330	18·997	482·516	2519	50	·02470	·3071

191	191·654	980·541	1172·195	·305	9·532	19·408	492·963	2469	49	·02529	·3134
192	192·665	979·835	1172·500	·305	9·738	19·827	503·600	2420	48	·02580	·3197
193	193·676	979·129	1172·805	·305	9·947	20·253	514·430	2372	47	·02632	·3264
194	194·686	978·424	1173·110	·305	10·160	20·687	525·455	2325	46	·02685	·3331
195	195·697	977·718	1173·415	·305	10·377	21·129	536·677	2279	45	·02740	·3399
196	196·708	977·012	1173·720	·305	10·597	21·579	548·098	2234	44	·02796	·3467
197	197·719	976·306	1174·025	·305	10·822	22·036	559·720	2190	43	·02853	·3535
198	198·730	975·600	1174·330	·305	11·051	22·502	571·545	2147	42	·02910	·3606
199	199·741	974·894	1174·635	·305	11·284	22·976	583·575	2105	41	·02967	·3677
200	200·753	974·187	1174·940	·305	11·521	23·458	595·812	2064	40	·03025	·3749
201	201·765	973·480	1175·245	·305	11·761	23·948	608·258	2024	39	·03083	·3821
202	202·777	972·773	1175·550	·305	12·006	24·446	620·915	1985	38	·03142	·3893
203	203·789	972·066	1175.855	·305	12·255	24·953	633·785	1953	37	·03201	·3968
204	204·801	971·359	1176·160	·305	12·508	25·468	646·873	1916	36	·03261	·4044
205	205·813	970·652	1176·465	·305	12·766	25·992	660·184	1880	36	·03323	·4120
206	206·825	969·945	1176·770	·305	13·028	26·525	673·723	1844	35	·03386	·4197
207	207·837	969·238	1177·075	·305	13·295	27·067	687·495	1809	34	·03450	·4275
208	208·849	968·531	1177·380	·305	13·568	27·619	701·505	1775	34	·03516	·4356
209	209·861	967·824	1177·685	·305	13·843	28·180	715·757	1741	33	·03584	·4440
210	210·874	967·116	1177·990	·305	14·122	28·751	730·254	1708	32	·03654	·4526
211	211·887	966·408	1178·295	·305	14·406	29·332	745·000	1676	32	·03725	·4615
212	212·900	965·700	1178·600	.305	14·696	29·9218	760·000	1644	31	·03797	·4705
213	213·913	964·992	1178·905	·008	14·991	30·522	775·234	1613	01	·03871	·4797
213·03	213·939	964·974	1178·913		15·000	30·540	775·705	1612		·03873	·4800

TABLE IV.

THE PROPERTIES OF SATURATED STEAM, AT EACH POUND OF PRESSURE, FROM ONE POUND TO TWO HUNDRED AND TEN POUNDS ON THE SQUARE INCH.

Gauge pressure, in lbs. on the square inch, with barometer at 29·922 inches.	Elastic Force.		Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.						Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	In lbs on the square inch.	In inches of mercury at 32°.		Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume.	Decrease of Volume.		
..	1	2·036	102·	102·086	24.354	1042·964	1145·050	7·400		20620·	9900·	·0030	·0374
..	2	4·072	126·266	126·440	15·437	1026·010	1152·450	4·681	2719	10720·	3394·	·0058	·0716
..	3	6 108	141·622	141·877	11·519	1015·254	1157·131	3·494	1187	7326·	1726·	·0085	·0623
..	4	8·144	153·070	153·396	9·326	1007·229	1160·625	2·824	670	5600·	1065·	·0112	·1385
..	5	10·180	162·330	162·722	7.855	1000·727	1163·449	2·377	447	4535·	721·	·0137	·1700
..	6	12·216	170·123	170·577	6·848	995·249	1165·826	2·070	307	3814·	514·	·0163	·2021
..	7	14·252	176 910	177·425	6·056	990·471	1167·896	1·830	240	3300·	390·	·0189	·2343
..	8	16·288	182·910	183·481	5·460	986·245	1169·726	1·649	181	2910·	303·	·0214	·2650
..	9	18·324	188·316	188·941	4·978	982·434	1171·375	1·502	147	2607·	247·	·0239	·2970
..	10	20·360	193·240	193·919	4·577	978·958	1172·877	1·381	121	2360·	203·	·0264	·3275
..	11	22·396	197·768	198·496	4·241	975·762	1174·258	1·279	102	2157·	169·	·0289	·3580
..	12	24·432	201·960	202·737	3·972	972·800	1175 537	1·197	82	1988·	142·	·0313	·3879
..	13	26·468	205 885	206·709	3·719	970·025	1176·734	1·121	76	1846·	124·	·0337	·4178
..	14	28·504	209·560	210·428	3·511	967·427	1177·855	1·057	64	1722·	110·	·0362	·4486
·304	15	30·540	213·025	213·939	3·313	964·973	1178·912	·997	60	1612·	98·	·0387	·4800
1·304	16	32·576	216·296	217·252	3·157	962·657	1179·909	·950	47	1514·	87·	·0413	·5107
2·304	17	34·612	219·410	220·409	3.010	960·450	1180·859	·905	45	1427·	76·4	·0437	·5414

3·304	18	36·648	222·378	223·419	2·866	958·345	1181·764	·864	41	1350·6	68·5	·0462	·5720
4·304	19	38·684	225·203	226·285	2·754	956·343	1182·628	·826	38	1282·1	61·8	·0487	·6026
5·304	20	40·720	227·917	229·039	2 637	954·415	1183·454	·792	34	1220·3	55 9	·0511	·6332
6·304	21	42·756	230·515	231·676	2·542	952·570	1184·246	·763	29	1164·4	50·9	·0536	·6637
7·304	22	44·792	233·017	234·218	2·454	950·791	1185·009	·735	28	1113·5	46·6	·0561	·6942
8·304	23	46·828	235·432	236·672	2·357	949·072	·744	·709	26	1066·9	42·8	·0585	·7247
9·304	24	48·864	237·752	239·029	2·285	947·424	1186·453	·686	23	1024·1	39·3	·0610	·7552
10·304	25	50·900	240·000	241·314	2·212	945·825	1187·139	·664	22	984·8	36·4	·0634	·7856
11·304	26	52·936	242·175	243·526	2·145	944·277	·803	·643	21	948·4	33·8	·0658	·8156
12·304	27	54·972	244·284	245·671	2·077	942·775	1188·446	·623	20	914·6	31·4	·0683	·8456
13·304	28	57·008	246·326	247·748	2·021	941·321	1189·069	·605	18	883·2	29·2	·0707	·8756
14·304	29	59·044	248·310	249·769	1 969	939·905	·674	·589	16	854·0	27·2	·0731	·9056
15·304	30	61·080	250·245	251·738	1·910	938·925	1190·263	·5728	16	826·8	25·6	·0755	·9355
16·304	31	63·115	252·122	253·648	1·864	937·1878	1190·8358	·5580	148	801·2	24·0	·0779	·9654
17·304	32	65·152	253·952	255·512	1·817	935·8818	1191·3938	·5440	140	777.2	22·5	·0803	·9953
18·304	33	67·187	255·735	257·329	1·774	934·6088	·9378	·5310	130	754·7	21·2	·0827	1·0251
19·304	34	69·224	257·476	259·103	1·732	933·3658	1192·4688	·5185	125	733·5	20·1	·0851	1 0549
20·304	35	71·259	259·176	260·835	1·692	932·1523	·9873	·5065	120	713·4	18·9	·0875	1·0846
21·304	36	73·296	260·835	262·527	1·655	930·9668	1193·4938	·4950	115	694·5	17·9	·0899	1·1142
22·304	37	75·331	262·458	264·182	1·619	929 8068	·9888	·4840	110	676·6	16·9	·0922	1·1438
23·304	38	77·378	264·045	265·801	1·585	928·6718	1194·4728	·4740	100	659·7	16·1	·0946	1·1733
24·304	39	79·403	265·599	267·386	1·552	927·5608	·9468	·4640	100	643·6	15·4	·0970	1·2027
25·304	40	81·440	267·120	268·938	1·522	926·4728	1195·4108	·4550	90	628·2	14·8	·0994	1·2320
26·804	41	83·477	268·611	270·460	1·494	925·4058	·8658	·4460	90	613 4	14·1	·1017	1·2612
27·304	42	85·512	270·073	271·954	1·463	924·3578	1196·3118	·4375	85	599·3	13·2	·1041	1·2904
28·304	43	87·547	271·507	273·417	1·438	923·3323	·7493	·4295	80	586·1	12·4	·1064	1·3196
29·304	44	89·584	272·915	274 855	1·411	922·3238	1197·1788	·4215	80	573·7	11·9	·1088	1·3487
30·304	45	91·620	274·296	276·266	1·385	921·3343	·6003	·4139	76	561·8	11·4	·1111	1·3767

TABLE IV.—*continued.*

Gauge Pressure, in lbs. on the square inch, with barometer at 29·922 inches.	Elastic Force.		Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.						Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	In lbs. on the square inch.	In inches of mercury at 32°.		Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume.	Decrease of Volume.		
31·304	46	93·656	275·652	277·651	1·365	920·3632	1198·0142	·4070	69	550.4	10·9	·1134	1·4057
32·304	47	95·692	276·986	279·016	1·339	919·4052	·4212	·4000	70	539·5	10·5	·1158	1·4347
33·304	48	97·729	278·297	280·355	1·317	918·4662	·8212	·3930	70	529·0	10·4	·1181	1·4636
34·304	49	99·764	279·585	281 672	1·297	917·5422	1199·2142	·3864	66	518·6	10·1	·1204	1·4925
35·304	50	101·801	280·854	282·969	1·274	916·6316	·6006	·3801	63	508·5	9·4	·1227	1·5213
36·304	51	103·836	282·099	284·243	1·256	915·7377	·9807	·3740	61	499·1	9 0	·1251	1·5501
37·304	52	105·873	283·326	285·499	1·237	914·8557	1200·3547	·3684	56	490·1	8·7	·1274	1·5788
38·304	53	107·908	284·534	286·736	1·216	913·9871	·7231	·3629	55	481·4	8·5	·1297	1·6074
39·304	54	109·945	285·724	287·952	1·201	913·1340	1201·0860	·3576	53	472·9	8·2	·1320	1·6360
40·304	55	111·980	286·897	289·153	1·182	912·2906	·4436	·3525	51	464·7	7·7	·1343	1·6645
41·304	56	114·027	288·052	290·335	1·168	911·4611	·7961	·3476	49	457·0	7·4	·1366	1·6930
42·304	57	116·053	289·112	291·503	1·151	910·6407	1202·1437	·3428	48	449·6	7·2	·1388	1·7214
43·304	58	118·089	290·316	292·654	1·136	909·8325	·4865	·3381	47	442·4	7·1	·1411	1.7498
44·304	59	120·125	291·425	293.790	1·121	909·0346	·8246	·3336	45	435·3	6·8	·1434	1·7782
45·304	60	122·160	292·520	294·911	1·105	908·2472	1203·1582	·3291	42	428·5	6·5	·1457	1·8055
46·304	61	124·196	293·598	296·016	1·092	907·4713	1203·4873	·3249	42	422·0	6·4	·14793	1·8337
47·304	62	126·232	294·663	297·108	1·077	906·7042	·8122	·3207	42	415·6	6·2	·15021	1·8618
48·304	63	128·268	295·714	298·185	1·064	905·9477	1204·1329	·3166	41	409·4	5·9	·15248	1·8899
49·304	64	130·304	396·752	299·249	1·051	905·2005	·4495	·3126	40	403·5	5·8	·15471	1·9179
50·304	65	132·340	297·777	300·300	1·038	904·4621	·7621	·3086	40	397·7	5·6	·15697	1·9459
51·304	66	134·376	298·789	301·338	1·026	903·7327	1205·0707	·3049	37	392·1	5·5	·15921	1·9737

52·304	67	136·412	299·789	302·364	1·013	903·0116	·8756	·3013	36	386·6	5·3	·16147	2·0014
53·304	68	138·448	300·776	303·377	1·003	902·2999	·6769	·2978	35	381·3	5·2	·16372	2·0291
54·304	69	140·484	301·753	304·380	·990	901·5947	·9747	·2944	34	376·1	4·9	·16598	2·0567
55·304	70	142·520	302·718	305·370	·980	900 8991	1206·2691	·2910	34	371·2	4·8	·16817	2·0842
56·304	71	144·556	303·673	306·350	·970	900·2101	·5601	·2879	31	366·4	4·7	·17038	2·1116
57·304	72	146·592	304·617	307·320	·959	899·5280	·8480	·2847	32	361·7	4·6	·17259	2·1390
58·304	73	148·628	305·551	308·279	·949	898·8537	1207·1327	·2816	31	357·1	4·5	·17481	2·1663
59·304	74	150·664	306·474	309·228	·938	898·1863	·4143	·2786	30	352·6	4·3	·17704	2·1936
60·304	75	152·700	307·388	310·166	·926	897·5269	·6929	·2755	31	348·3	4·2	·17923	2·2209
61·304	76	154·736	308·290	311·092	·919	896·8764	1207·9684	·2727	28	344·1	4·1	·18142	3·2481
62·304	77	156·772	309·184	312·011	·909	896·2301	1208·2411	·2699	28	340·0	4·0	·18360	2·2753
63·304	78	158·808	310·069	312·920	·901	895·5910	·5110	·2671	28	336·0	3·9	·18579	2·3025
64·304	79	160·844	310·945	313·821	·891	894·9571	·7781	·2643	28	332·1	3·8	·18797	2·3296
65·304	80	162·880	311·812	314·712	·883	894·3304	1209·0424	·2618	25	328·3	3·7	·19015	2·3567
66·304	81	164·916	312·670	315·595	·873	893·7092	·3042	·2592	26	324·6	3·7	·19232	2·3838
67·304	82	166·952	313·520	316·468	·865	893·0954	·5634	·2567	25	320·9	3·6	·19454	2·4108
68·304	83	168·988	314·361	317·333	·857	892·4871	·8201	·2542	25	317·3	3·4	·19674	2·4378
69·304	84	171·024	315·195	318·190	·850	891·8843	1210·0743	·2519	23	313·9	3·4	·19887	2·4647
70·304	85	173·060	316·021	319·040	·842	891·2862	·3262	·2496	23	310·5	3·3	·20105	2·4916
71·304	86	175·096	316·839	319·882	·835	890·6938	·5758	·2473	23	307·2	3·2	·20321	2·5184
72·304	87	177·132	317·650	320·717	·826	890·1061	·8231	·2450	23	304·0	3·2	·20535	2·5452
73·304	88	179·168	318·453	321·543	·819	889·5251	1211·0681	·2429	21	300·8	3·1	·20753	2·5719
74·304	89	181·204	319·249	322·362	·814	888·9490	·3110	·2408	21	297·7	3·0	·20970	2·5987
75·304	90	185·240	320·089	323·176	·805	888 3758	·5518	·2386	21	294·7	2·9	·21183	2·6249
76·304	91	185·276	320·821	323·981	·800	887·8094	1211·7904	·2366	20	291·8	2·9	·21393	2·6514
77·304	92	187·312	321·597	324·781	·791	887·2460	1212·0270	·2346	20	288·9	2·8	·21608	2·6778
78·304	93	189·348	322·366	325·572	·786	886·6896	·2616	·2326	20	286·1	2·8	·21829	2·7041
79·304	94	191·384	323·128	326·358	·778	886·1362	·4942	·2305	21	283·3	2·7	·22045	2·7304

Table IV.—*continued.*

Gauge Pressure, in lbs. on the square inch, with barometer at 29·922 in.	Elastic Force.		Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.						Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
	In lbs. on the square inch.	In inches of mercury at 32°.		Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume.	Decrease of Volume.		
80·304	95	193·422	323·884	327·136	·773	885·5887	·7247	·2287	18	280·6	2·6	·22247	2·7566
81·304	96	195·456	324·634	327·909	·766	885·0444	·9534	·2268	19	278·0	2·6	·22455	2·7828
82·304	97	197·492	325·378	328·675	·758	884·5052	1213·1802	·2249	19	275·4	2·6	·22667	2·8089
83·304	98	199·528	326·114	329·433	·753	883·9721	·4051	·2230	19	272·8	2·5	·22883	2·8350
84·304	99	201·564	326·845	330·186	·749	883·4421	·6281	·2213	17	270·3	2·4	·23095	2·8610
85·304	100	203·600	327·571	330·935	·743	882·9144	·8494	·2195	18	267·9	2·4	·23302	2·8870
86·304	101	205·636	328·291	331·678	·736	882·3909	1214·0689	·2178	17	265·5	2·3	·23510	2·9129
87·304	102	207·672	329·005	332·414	·731	881·8727	·2867	·2160	18	263·2	2·3	·23717	2·9388
88·304	103	209·708	329·714	333·145	·724	881·3577	·5027	·2144	16	260·9	2·2	·23925	2·9646
89·304	104	211·744	330·416	333·869	·718	880·8481	·7171	·2128	16	258·7	2·2	·24132	2·9904
90·304	105	213·780	331·113	334·587	·714	880·3429	·9299	·2111	17	256·5	2·2	·24340	3·0161
91·304	106	215·816	331·805	335·301	·708	879·8400	1215·1410	·2096	15	254·3	2·1	·24547	3·0418
92·304	107	217·852	332·492	336·009	·705	879·3416	·3506	·2081	15	252·2	2·1	·24754	3·0675
93·304	108	219·888	333·174	336·714	·697	878·8447	·5587	·2065	16	250·1	2·1	·24961	3·0931
94·304	109	221·924	333·851	337·411	·694	878·3542	·7652	·2051	14	248·0	2·0	·25168	3·1187
95·304	110	223·960	334·523	338·105	·690	877·8653	·9703	·2036	15	246·0	2·0	·25376	3·1443
96·304	111	225·996	335·191	338·795	·684	877·3789	1216·1739	·2021	15	244·0	2·0	·25582	3·1699
97·304	112	228·032	335·854	339·479	·678	876·8970	·3760	·2008	13	242 0	1·9	·25788	3·1954
98·304	113	230·068	336·511	340·157	·675	876·4198	·5768	·1994	14	240·1	1·9	·25994	3·2209
99·304	114	232·104	337·165	440·832	·670	875·9442	·7762	·1979	15	238·2	1·9	·26199	3·2464
100·304	115	234·140	337·814	341·502	·667	875·4721	·9741	·1967	12	236·3	1·8	·26405	3·2718

101·304	116	236·176	338·459	342·169	·662	875·0018	1217·1708	·1954	13	234·5	1·8	·26611	3·2972
102·304	117	238·212	339·100	342·831	·657	874·5352	·3662	·1940	14	232·7	1·7	·26816	3·3225
103·304	118	240·248	339·736	343·488	·653	874·0722	·5602	·1928	12	231·0	1·7	·27020	3·3478
104·304	119	242·284	340·368	344·141	·648	873·6120	·7530	·1915	13	229·3	1·7	·27224	3·3730
105·304	120	244·320	340·995	344·789	·643	873·1555	·9445	·1902	13	227·6	1·6	·27428	3·3988
106·304	121	246·356	341·618	345·432	·641	872·7027	1218·1347	·1891	11	226·	1·6	·27628	3·4238
107·304	122	248·392	342·238	346·073	·636	872·2508	·3238	·1879	12	224·4	1·6	·27828	3·4480
108·304	123	250·428	342·854	346·709	·634	871·8027	·5117	·1866	13	222·8	1·6	·28027	3·4729
109·304	124	252·464	343·466	347·343	·629	871·3553	·6983	·1855	11	221·2	1·5	·28227	3·4978
110·304	125	254·500	344·074	347·972	·624	870·9118	·8838	·1843	12	219·7	1·5	·28426	3·5226
111·304	126	256·536	344·678	348·596	·621	870·4721	1219·0681	·1831	12	218·2	1·5	·28625	3·5474
112·304	127	258·572	345·279	349·217	·618	870·0342	·2512	·1821	10	216·7	1·5	·28824	3·5721
113·304	128	260·608	345·876	349·835	·613	869·5983	·4333	·1810	11	215·2	1·5	·29023	3·5967
114·304	129	262·644	346·459	350·448	·611	869·1663	·6143	·1798	12	213·7	1·4	·29222	3·6213
115·304	130	264·680	347·059	351·059	·606	868·7351	·7941	·1788	10	212·3	1·4	·29420	3·6458
116·304	131	266·716	347·644	351·665	·602	868·3079	·9729	·1777	11	210·9	1·4	·29618	3·6703
117·304	132	268·752	348·227	352·267	·600	867.8836	1220·1506	·1765	12	209·5	1·4	·29816	3·6947
118·304	133	270·788	348·806	352·867	·596	867·4601	·3271	·1756	9	208·1	1·4	·30013	3·7190
119·304	134	272·824	349·382	353·463	·592	867·0397	·5027	·1746	10	206·7	1·3	·30209	3·7433
120·304	135	274·860	349·954	354·055	·589	866·6223	·6773	·1735	11	205·4	1·3	·30405	3·7676
121·304	136	276·896	350·523	354·644	·586	866·2068	1220·8508	·1726	9	204·1	1·3	·30601	3·7921
122·304	137	278·932	351·089	355·230	·583	866·7934	1221·0234	·1716	10	202·8	1·3	·30796	3·8166
123·304	138	280·968	351·752	355·813	·579	865·3820	·1950	·1705	11	201·5	1·3	·30990	3·8412
124·304	139	283·004	352·211	356·392	·577	864·9735	·3655	·1696	9	200·2	1·2	·31186	3·8658
125·304	140	285·040	352·767	356·969	·572	864·5661	·5351	·1686	10	199·0	1·2	·31386	3·8904
126·364	141	287·076	353·319	357·541	·569	864·1627	·7037	·1676	10	197·8	1·2	·31587	3·9151
127·304	142	289·112	353·869	358·110	·567	863·7613	·8713	·1668	8	196·6	1·2	·31788	3·9398
128·304	143	291·148	354·416	358·677	·563	863·3611	1222·0381	·1659	9	195·4	1·2	·31990	3·9646

TABLE IV.—*continued.*

Gauge pressure, in lbs. on the square inch, with barometer at 29·922 inches.	Elastic Force. In lbs. on the square inch.	Elastic Force. In inches of mercury at 32°.	Temperature in dgrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit. Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference	2nd Difference.	Volume: that of an equal weight of water being 1; known as specific volume. Volume.	Decrease of Volume.	Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
129·304	144	293·184	354·960	359·240	·561	862·9640	·2040	·1649	10	194·2	1·2	·32190	3·9894
130·304	145	295·220	355·501	359·801	·558	862·5679	·3689	·1641	8	193·0	1·1	·32391	4·0143
131·304	146	297·256	356·039	360·359	·554	862·1740	·5330	·1632	9	191·9	1·1	·32592	4·0392
132·304	147	299·292	356·574	360·913	·552	861·7832	·6962	·1622	10	190·8	1·1	·32794	4·0641
133·304	148	301·328	357·106	361·465	·548	861·3934	·8584	·1614	8	189·7	1·1	·32995	4·0891
134·304	149	303·364	357·635	362·013	·546	861·0068	1223·0198	·1605	9	188·6	1·1	·33196	4·1139
135·304	150	305·400	358·161	362·559	·541	860·6213	·1803	·1596	9	187·5	1·1	·33400	4·1388
136·304	151	307·436	358·683	363·100	·540	860·2399	1223·3399	·1589	7	186·4	1·1	·33580	4·1618
137·304	152	309 472	358·203	363·640	·537	859 8588	·4988	·1581	8	185·3	1·	·33761	4·1848
138·304	153	311·508	359·721	364·177	·534	859·4799	·6569	·1572	9	184·3	1·	·33942	4·2078
139·304	154	313·544	360·236	364·711	·532	859·1031	·8141	·1565	7	183·3	1·	·34123	4·2307
140·304	155	315·580	360·749	365.243	·530	858·7276	·9706	·1557	8	182·3	1·	·34304	4·2536
141·304	156	317·616	361·260	365·773	·527	858·3533	1224·1263	·1548	9	181·3	1·	·34485	4·2764
142·304	157	319·652	361·768	366·300	·524	857·9811	·2811	·1541	7	180·3	1·	·34666	4·2992
143·304	158	321·688	362·273	366·824	·523	857·6112	·4352	·1533	8	179·3	1·	·34847	4·3219
144·304	159	323·724	362·776	367·347	·520	857·2415	·5885	·1525	8	178·3	1·	·35028	4·3446
145·304	160	325·760	363·277	367·867	·516	856·8740	·7410	·1519	6	177·3	·9	·35209	4·3672
146·304	161	327·796	363·774	368·383	·515	856·5099	·8929	·1512	7	176·4	·9	·35397	4·3898
147·304	162	329·832	364·270	368·898	·512	856·1461	1225·0441	·1505	7	175·5	·9	·35585	4·4123
148·304	163	331·868	364·764	369·410	·510	855·7846	·1946	·1497	8	174·6	·9	·35773	4·4348
149·304	164	333·904	365·255	369·920	·508	855·4243	·3443	·1491	6	173·7	·9	·35961	4·4572

150·304	165	335·940	365·744	370·428	·506	855·0654	·4934	·1483	8	172·8	·9	·36149	4·4794
151·304	166	337·776	366·232	370·934	·504	854·7077	1225·6417	·1477	6	171·9	·9	·36337	4·5020
152·304	167	340·012	366·717	371·438	·501	854·3514	·7894	·1470	7	171·0	·9	·36525	4·5246
153·304	168	342·048	367·199	371·939	·498	853·9974	·9364	·1462	8	170·1	·9	·36714	4·5473
154·304	169	344·084	367·680	372·437	·497	853·6456	1226·0826	·1456	6	169·2	·9	·36903	4·5700
155·304	170	346·120	368·158	372·934	·493	853·2942	·2282	·1449	7	168·4	·8	·37092	4·5927
156·304	171	348·156	368·632	373·427	·491	852·9461	·3731	·1444	5	167·6	·8	·37272	4·6154
157·304	172	350·192	369·105	373·918	·490	852·5995	·5175	·1438	6	166·8	·8	·37452	4·6381
158·304	173	352·228	369·576	374·408	·487	852·2533	·6613	·1431	7	166·0	·8	·37632	4·6608
159·304	174	354·264	370·045	374·895	·485	851·9094	·8044	·1426	5	165·2	·8	·37812	4·6835
160·304	175	356·300	370·512	375 380	·485	851·5670	·9470	·1419	7	164·4	·8	·37992	4·7062
161·304	176	358·336	370·978	375·865	·482	851·2239	1227·0889	·1414	5	163·6	·8	·38172	4·7289
162·304	177	360·372	371·442	376·347	·480	850·8833	·2303	·1408	6	162·8	·8	·38353	4·7516
163·304	178	362·408	371·904	376·827	·478	850·5441	·3711	·1401	7	162·0	·8	·38534	4·7743
164·304	179	364·444	372·364	377·305	·476	850·2062	·5112	·1396	5	161·2	·8	·38715	4·7970
165·304	180	366·480	372·822	377·781	·474	849·8698	·6508	·1389	7	160·4	·7	·38895	4·8197
166·304	181	368·516	373·275	378·255	·472	849·5347	1227·7897	·1384	5	159·7	·7	·39077	4·8421
167·304	182	370·552	373·731	378·727	·470	849·2011	·9281	·1378	6	159·0	·7	·39259	4·8645
168·304	183	372·588	374·183	379·197	·468	848·8689	1228·0659	·1371	7	158·3	·7	·39441	4·8869
169·304	184	374·624	374·633	379·665	·466	848·5380	·2030	·1366	5	157·6	·7	·39624	4·9093
170·304	185	376·660	375·081	380·131	·464	848·2086	·3396	·1359	7	156·9	·7	·39807	4·9317
171·304	186	378·696	375·527	380·595	·461	847·8805	·4755	·1354	5	156 2	·7	·39990	4·9541
172·304	187	380·732	375·971	381·056	·460	847·5549	·6109	·1348	6	155·5	·7	·40173	4·9765
173·304	188	382·768	376·413	381·516	458	847·2297	·7457	·1341	7	154·8	·7	·40356	4·9989
174·304	189	384·804	376·853	381·974	·455	846·9058	·8798	·1336	5	154·1	·7	·40539	5·0213
175·304	190	386·840	377·291	382·429	·454	846·5844	1229·0134	·1329	7	153·4	·7	·40722	5·0437
176·304	191	388·876	377·727	382·883	·452	846·2633	·1463	·1324	5	152·7	·7	·40899	5·0661
177·304	192	390·912	378·161	383·335	·450	845·9437	·2787	·1319	5	152·0	·7	·41076	5·0885

TABLE IV.—*continued.*

Gauge pressure, in lbs. on the square inch, with barometer at 29·922 inches.	Elastic Force. In lbs. on the square inch.	Elastic Force. In inches of mercury at 32°.	Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit. Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume: that of an equal weight of water being 1; known as specific volume. Volume.	Decrease of Volume.	Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
178·304	193	392·948	378·593	383·785	·448	845.6256	·4106	·1313	6	151·3	·6	·41253	5·1109
179·304	194	394·984	379·023	384·233	·446	845·3089	·5419	·1309	4	150·7	·6	·41430	5·1333
180·304	195	397·020	379·452	384·679	·444	844·9938	·6728	·1303	6	150·1	·6	·41607	5·1557
181·304	196	399·056	379·979	385·123	·444	844·6801	1229·8031	·1299	4	149·5	·6	·41784	5·1780
182·304	197	401·092	380·305	385·567	·441	844·3660	·9330	·1293	6	148·9	·6	·41962	5·2002
183·304	198	403·128	380·729	386·008	·441	844·0543	1230·0623	·1289	4	148·3	·6	·42140	5·2223
184·304	199	405·164	381·152	386·449	·438	843·7422	·1912	·1284	5	147·7	·6	·42318	5·2443
185·304	200	407·200	381·573	386·887	·437	843·4326	·3196	·1278	6	147·1	·6	·42496	5·2662
186·304	201	409·236	381·992	387·324	·436	843·1234	·4474	·1274	4	146·5	·6	·42667	5·2880
187·304	302	411·272	382·410	387·760	·434	842·8148	·5748	·1268	6	145·9	·6	·42838	5·3097
188·304	203	413·308	382·827	388·194	·433	842·5076	·7016	·1264	4	145·3	·6	·43009	5·3313
189·304	204	415·344	383·242	388·627	·430	842·2010	·8280	·1259	5	144·7	·6	·43180	5·3528
190·304	205	417·380	383·655	389·057	·428	841·8969	·9539	·1253	6	144·1	·6	·43351	5·3742
191·304	206	419·416	384·066	389·485	·427	841·5942	1231·0792	·1249	4	143·5	·6	·43523	5·3955
192·304	207	421·452	384·475	389·912	·425	841·2921	·2041	·1243	6	142·9	·6	·43695	5·4167
193·304	208	423·488	384·883	390·337	·422	840·9914	·3284	·1239	4	142·3	·5	·43866	5·4378
194·304	209	425·524	385·288	390·759	·420	840·6933	·4523	·1234	5	141·8	·5	·44039	5·4582
195·304	210	427.560	385·671	391·179		840·3967	·5757		..	141·3		·44211	5·4784

TABLE V.

THE PROPERTIES OF SATURATED STEAM, AT PRESSURES FROM ONE ATMOSPHERE TO EIGHTEEN ATMOSPHERES.

Elastic Force.			Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit.						Volume: that of an equal weight of water being 1; known as specific volume.		Weight of one cubic foot, in decimals of of a pound.	Specific Gravity; the atmosphere at 32° being 1.
In Atmospheres.	In pounds on the square inch.	In inches of mercury at 32°.		Number contained in the water.	Difference.	Number required for evaporation known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume.	Decrease of Volume.		
·25	3·674	7·480	149·649	149·943	29·598	1009·6372	1159·5802	8·9548		6054·	2895·	·01032	·1279
·5	7·348	14·961	179·008	179·541	19·176	988·9940	1168·5350	5·7910	3·1638	3159·	1012·	·01976	·2449
·75	11·022	22·441	197·987	198·717	14·183	975·6090	1174·3260	4·2740	1·5170	2147·	503·	·02912	·3608
1·	14·696	29·922	212 000	212·900	21·425	965·7000	1178·6000	6·4425		1644·	532·5	·03797	·4705
1·5	22·044	44·883	233·123	234·325	16·213	950·7175	1185·0425	4·8632	1·5793	1111·5	269·5	·05616	·6959
2·	29·392	59 844	249·068	250·538	13·212	939·3677	1189·9057	3·9553	·9079	842·0	160·9	·07415	·9188
2·5	36·740	74·805	262·036	263·750	11 228	930·1110	1193·8610	3·3553	·6000	681·1	108·4	·09166	1·1358
3·	44·088	89·766	273·037	274·978	9·811	922·2383	1197·2163	2·9271	·4282	572·7	77·5	·10894	1·3500
3·5	51·436	104·727	282·634	284·789	8·753	915·3544	1200·1434	2·6080	·3191	495·2	58·3	·12610	1·5613
4·	58·784	119·688	291·185	293·542	7·931	909·2094	1202·7514	2·3595	·2485	436·9	45·5	·14284	1·7700
4·5	66·132	134·649	298·921	301·473	7·260	903·6379	1205·1109	2·1585	·2010	391·4	36·5	·15949	1·9763
5·	73·480	149·610	305·994	308·733	6·708	898·5364	1207·2694	1·9898	·1687	354·9	29·7	·17584	2·1790
5·5	80·828	164·571	312·522	315·441	6·244	893·8182	1209·2592	1·8517	·1381	325·2	24·8	·19195	2·3778
6·	88·176	179·532	318·593	321·685	5·854	889·4259	1211·1109	1·7335	·1182	300·4	21·1	·20780	2·5750

TABLE V.—*continued.*

Elastic Force. In Atmospheres.	Elastic Force. In pounds on the square inch.	Elastic Force. In inches of mercury at 32°.	Temperature in degrs. Fahrenheit of the steam, and of the water from which it is evaporated.	Number of British Thermal Units contained in one pound, reckoned from zero Fahrenheit. Number contained in the water.	Difference.	Number required for evaporation, known as latent heat, or heat of vaporisation.	Total number contained in the steam.	Difference.	2nd Difference.	Volume: that of an equal weight of water being 1; known as specific volume. Volume.	Decrease of Volume.	Weight of one cubic foot, in decimals of a pound.	Specific Gravity; the atmosphere at 32° being 1.
6·5	95·524	194·493	324·277	327·539	5·509	885·3054	1212·8444	1·6306	·1029	279·3	18·1	·22350	2·7695
7·	102·872	209·454	329·623	333·048	5·206	881·4270	1214·4750	1·5393	·0913	261·2	15·6	·23899	2·9615
7·5	110·220	224·415	334·670	338·254	4·947	877·7603	1216·0143	1·4610	·0783	245·6	13·6	·25417	3·1500
8·	117·568	239·376	339·461	343·201	4·715	874·2743	1217·4753	1·3914	·0696	232·0	12·1	·26906	3·3340
8·5	124·916	254·337	344·023	347·916	4·509	871·9507	1218·8667	1·3292	·0622	219·9	10·8	·28387	3·5176
9·	132·264	269·298	348·380	352·425	4·318	867·7709	1220·1959	1·2722	·0570	209·1	9·6	·29852	3·7000
9·5	139·612	284·259	352·551	356·743	4·145	864·7251	1221·4681	1·2205	·0517	199·5	8·7	·31290	3·8773
10·	146·960	299·220	356·553	360·888	7·830	861·8006	1222·6886	2·3019		190·8	14·9	·32718	4·0543
11·	161·656	329·142	364·099	368·718	7·314	856·2725	1224·9905	2·1475	·1544	175·9	12·6	·35494	4·4000
12·	176·352	359·064	371·141	376·032	6·870	851·1060	1227·1380	2·0151	·1324	163·3	10·6	·38227	4·7370
13·	191·048	388·986	377·748	382·902	7·468	846·2511	1229·1531	1·8947	·1204	152·7	9·0	·40880	5·0657
14·	205·744	418·908	383·960	389·370	6·136	841·6778	1231·0478	1·7934	·1013	143·7	7·8	·43441	5·3830
15·	220·440	448·830	389·840	395·506	5·834	837·3352	1232·8412	1·7046	·0888	135·9	6·9	·45934	5·6920
16·	235·136	478·752	395·429	401·340	5·572	833·2058	1234·5458	1·6245	·0801	129·0	6·3	·48390	6·0000
17·	249·832	508·674	400·755	406·912	5·334	829·2583	1236·1703	1·5515	·0730	122·7	5·9	·50875	6·3036
18·	264·528	538·596	405·842	412·246		825·4758	1237·7218			116·8		·53446	6·6227

TABLE VI.

HYPERBOLIC LOGARITHMS.

IN estimating the power which an engine will exert with a given pressure of steam, to be cut off at any given point of the stroke, we ascertain the mean pressure on the square inch which will be exerted during the stroke by means of the Table of Hyperbolic Logarithms, which latter are calculated for expansion according to the law of Mariotte.

RULE.—Divide the length of the stroke by the length of the space into which the steam is admitted; find in the Table the logarithm of the quotient. If the quotient is not in the Table, its logarithm will be found by adding the relative difference to the logarithm of the nearest smaller number; as, for example, the logarithm of 1·28 is ·223 + ·024 = ·247. Then find the terminal pressure, by dividing the initial pressure by the proportion of the stroke during which the steam is admitted, and muptiply it by the logarithm + 1 found as above; the product will be the mean pressure through the stroke.

Example.—Suppose the length of the stroke to be 48 inches, the initial pressure to be 80 lbs. per square inch, and the steam to be cut off at 12 inches of the stroke, what will be the mean pressure?

$$48 \div 12 = 4. \quad \text{Hyp. log. of } 4 = 1 \cdot 386 + 1 = 2 \cdot 386.$$

Then, $80 \div 4 = 20 \times 2 \cdot 386 = 47 \cdot 72$ lbs., the mean pressure required.

The pressures given above are the total pressures, measured from perfect vacuum. To find the initial pressure, add the atmospheric pressure to the pressure shown by the gauge, and from the mean pressure found as above subtract the counter-pressure, to ascertain the effective mean pressure exerted. Thus, in the above case, the gauge is supposed to show a pressure of 65 lbs. only, and if the calculation is being made for a condensing engine, the estimated loss from imperfect vacuum must be subtracted, and if for a non-condensing engine, the pressure of the atmosphere, and also any estimated counter-pressure above that, must be subtracted, from 47·72 lbs., the mean pressure obtained by the calculation.

TABLE OF HYPERBOLIC LOGARITHMS.

Numb.	Log.	Numb.	Log.	Numb.	Log.	Numb.	Log.
1·05	·049	2·55	·936	4·05	1·399	5·55	1·714
1·1	·095	2·6	·956	4·1	1·411	5·6	1·723
1·15	·140	2·65	·975	4·15	1·423	5·65	1·732
1·2	·182	2·7	·993	4·2	1·435	5·7	1·740
1·25	·223	2·75	1·012	4·25	1·447	5·75	1·749
1·3	·262	2·8	1·030	4·3	1·459	5·8	1·758
1·35	·300	2·85	1·047	4·35	1·470	5·85	1·766
1·4	·336	2·9	1·065	4·4	1·482	5·9	1·775
1·45	·372	2·95	1·082	4·45	1·493	5·95	1·783
1·5	·405	3·0	1·099	4·5	1·504	6·0	1·792
1·55	·438	3·05	1·115	4·55	1·515	6·05	1·800
1·6	·470	3·1	1·131	4·6	1·526	6·1	1·808
1·65	·500	3·15	1·147	4·65	1·537	6·15	1·816
1·7	·531	3·2	1·163	4·7	1·548	6·2	1·824
1·75	·560	3·25	1·179	4·75	1·558	6·25	1·833
1·8	·588	3·3	1·194	4·8	1·569	6·3	1·841
1·85	·615	3·35	1·209	4·85	1·579	6·35	1·848
1·9	·642	3·4	1·224	4·9	1·589	6·4	1·856
1·95	·668	3·45	1·238	4·95	1·599	6·45	1·864
2·0	·693	3·5	1·253	5·0	1·609	6·5	1·872
2·05	·718	3·55	1·267	5·05	1·619	6·55	1·879
2·1	·742	3·6	1·281	5·1	1·629	6·6	1·887
2·15	·765	3·65	1·295	5·15	1·639	6·65	1·895
2·2	·788	3·7	1·308	5·2	1·649	6·7	1·902
2·25	·811	3·75	1·322	5·25	1·658	6·75	1·910
2·3	·833	3·8	1·335	5·3	1·668	6·8	1·917
2·35	·854	3·85	1·348	5·35	1·677	6·85	1·924
2·4	·875	3·9	1·361	5·4	1·686	6·9	1·931
2·45	·896	3·95	1·374	5·45	1·696	6·95	1·939
2·5	·916	4·0	1·386	5·5	1·705	7·0	1·946

TABLE OF HYPERBOLIC LOGARITHMS—*continued.*

Numb.	Log.	Numb.	Log.	Numb.	Log.	Numb.	Log.
7·05	1·953	8·05	2·086	9·05	2·203	15·	2·708
7·1	1·960	8·1	2·092	9·1	2·208	20·	2·996
7·15	1·967	8·15	2·098	9·15	2·214	25·	3·219
7·2	1·974	8·2	2·104	9·2	2·219	30·	3·401
7·25	1·981	8·25	2·110	9·25	2·225	35·	3·555
7·3	1·988	8·3	2·116	9·3	2·230	40·	3·689
7·35	1·995	8·35	2·122	9·35	2·235	45·	3·807
7·4	2·001	8·4	2·128	9·4	2·241	50·	3·912
7·45	2·008	8·45	2·134	9·45	2·246	55·	4·007
7·5	2·015	8·5	2·140	9·5	2·251	60·	4·094
7·55	2·022	8·55	2·146	9·55	2·257	65·	4·174
7·6	2·028	8·6	2·152	9·6	2·262	70·	4·248
7·65	2·035	8·65	2·158	9·65	2·267	75·	4·317
7·7	2·041	8·7	2·163	9·7	2·272	80·	4·382
7·75	2·048	8·75	2·169	9·75	2·277	85·	4·443
7·8	2·054	8·8	2·175	9·8	2·282	90·	4·500
7·85	2·061	8·85	2·180	9·85	2·287	95·	4·554
7·9	2·067	8·9	2·186	9·9	2·293	100·	4·605
7·95	2·073	8·95	2·192	9·95	2·298	1000	6·908
8·0	2·079	9·0	2·197	10·0	2·303	10,000	9·210

Table VII.

Areas of Circles, in Square Inches, from $\frac{1}{16}$ in. to 4 ins. diameter, varying by sixteenths; and from 4 ins. to 100 ins. diameter, varying by quarter-inches.

Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.
$\frac{1}{16}$	·00307	$2\frac{7}{16}$	4·6664	6·75	35·78
$\frac{1}{8}$	·01227	$2\frac{1}{2}$	4·9087	7·	38·48
$\frac{3}{16}$	·02761	$2\frac{9}{16}$	5·1573	7·25	41·28
$\frac{1}{4}$	·04909	$2\frac{5}{8}$	5·4119	7·5	44·17
$\frac{5}{16}$	·07670	$2\frac{11}{16}$	5·6727	7·75	47·17
$\frac{3}{8}$	·11045	$2\frac{3}{4}$	5·9396	8·	50·26
$\frac{7}{16}$	·15033	$2\frac{13}{16}$	6·2126	8·25	53·45
$\frac{1}{2}$	·19635	$2\frac{7}{8}$	6·4918	8·5	56·74
$\frac{9}{16}$	·24850	$2\frac{15}{16}$	6·7772	8·75	60·13
$\frac{5}{8}$	·30680	3	7·0686	9·	63·61
$\frac{11}{16}$	·37122	$3\frac{1}{16}$	7·3662	9·25	67·19
$\frac{3}{4}$	·44179	$3\frac{1}{8}$	7·6699	9·5	70·88
$\frac{13}{16}$	·51849	$3\frac{3}{16}$	7·9798	9·75	74·66
$\frac{7}{8}$	·60132	$3\frac{1}{4}$	8·2958	10·	78·54
$\frac{15}{16}$	·69029	$3\frac{5}{16}$	8·6179	10·25	82·51
1	·78540	$3\frac{3}{8}$	8·9462	10·5	86·59
$1\frac{1}{16}$	·88664	$3\frac{7}{16}$	9·2806	10·75	90·76
$1\frac{1}{8}$	·99402	$3\frac{1}{2}$	9·6211	11·	95·03
$1\frac{3}{16}$	1·1075	$3\frac{9}{16}$	9·9678	11·25	99·40
$1\frac{1}{4}$	1·2272	$3\frac{5}{8}$	10·321	11·5	103·86
$1\frac{5}{16}$	1·3530	$3\frac{11}{16}$	10·680	11·75	108·38
$1\frac{3}{8}$	1·4849	$3\frac{3}{4}$	11·045	12·	113·10
$1\frac{7}{16}$	1·6230	$3\frac{13}{16}$	11·416	12·25	117·86
$1\frac{1}{2}$	1·7671	$3\frac{7}{8}$	11·793	12·5	122·72
$1\frac{9}{16}$	1·9175	$3\frac{15}{16}$	12·177	12·75	127·68
$1\frac{5}{8}$	2 0739			13·	132·73
$1\frac{11}{16}$	2·2365			13·25	137·89
$1\frac{3}{4}$	2·4053	4·	12·56	13·5	143·14
$1\frac{13}{16}$	2·5802	4·25	14·18	13·75	148·49
$1\frac{7}{8}$	2·7612	4·5	15·90	14·	153·94
$1\frac{15}{16}$	2·9483	4·75	17·71	14·25	159·49
2	3·1416	5·	19·63	14·5	165·13
$2\frac{1}{16}$	3·3410	5·25	21·64	14·75	170·87
$2\frac{1}{8}$	3·5466	5·5	23·75	15·	176·71
$2\frac{3}{16}$	3·7583	5·75	25·96	15·25	182·65
$2\frac{1}{4}$	3·9761	6·	28·27	15·5	188·69
$2\frac{5}{16}$	4·2001	6·25	30·67	15·75	194·83
$2\frac{3}{8}$	4·4301	6·5	33·17	16·	201·06

TABLE VII.—*continued.*

Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.
16·25	207·39	27·25	583·21	38·25	1149·09
16·5	213·83	27·5	593·96	38·5	1164·16
16·75	220·35	27·75	604·81	38·75	1179·33
17·	226·98	28·	615·75	39·	1194·59
17·25	233·71	28·25	626·80	39·25	1209·96
17·5	240·53	28·5	637·94	39·5	1225·42
17·75	247·45	28·75	649·18	39·75	1240·98
18·	254·47	29·	660·52	40·	1256·64
18·25	261·59	29·25	671·96	40·25	1272·40
18·5	268·80	29·5	683·49	40·5	1288·25
18·75	276·12	29·75	695·13	40·75	1304·21
19·	283·53	30·	706·86	41·	1320·26
19·25	291·04	30·25	718·69	41·25	1336·41
19·5	298·65	30·5	730·62	41·5	1352·66
19·75	306·36	30·75	742·64	41·75	1369·00
20·	314·16	31·	754·77	42·	1385·44
20·25	322·06	31·25	766·99	42·25	1401·99
20·5	330·06	31·5	779·31	42·5	1418·63
20·75	338·16	31·75	791·73	42·75	1435·36
21·	346·36	32·	804·25	43·	1452·20
21·25	354·66	32·25	816·87	43·25	1469·14
21·5	363·05	32·5	829·58	43·5	1486·20
21·75	371·54	32·75	842·39	43·75	1503·30
22·	380·13	33·	855·30	44·	1520·53
22·25	388·82	33·25	868·31	44·25	1537·86
22·5	397·61	33·5	881·42	44·5	1555·29
22·75	406·49	33·75	894·62	44·75	1572·81
23·	415·48	34·	907·92	45·	1590·44
23·25	424·56	34·25	921·32	45·25	1608·16
23·5	433·74	34·5	934·82	45·5	1625·97
23·75	443·01	34·75	948·42	45·75	1643·89
24·	452·39	35·	962·12	46·	1661·91
24·25	461·86	35·25	975·91	46·25	1680·02
24·5	471·44	35·5	989·80	46·5	1698·23
24·75	481·11	35·75	1003·79	46·75	1716·54
25·	490·87	36·	1017·88	47·	1734·95
25·25	500·74	36·25	1032·06	47·25	1753·45
25·5	510·71	36·5	1046·35	47·5	1772·06
25·75	520·77	36·75	1060·73	47·75	1790·76
26·	530·93	37·	1075·22	48·	1809·56
26·25	541·19	37·25	1089·79	48·25	1828·41
26·5	551·55	37·5	1104·47	48·5	1847·26
26·75	562·00	37·75	1119·24	48·75	1866·26
27·	572·56	38·	1134·12	49·	1884·96

TABLE VII.—*continued.*

Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.
49·25	1905·03	60·25	2851·0	71·25	3987·1
49·5	1924·23	60·5	2874·7	71·5	4015·1
49·75	1943·86	60·75	2898·5	71·75	4043·2
50·	1963·50	61·	2922·4	72·	4071·5
50·25	1983·18	61·25	2946·4	72·25	4099·8
50·5	2002·77	61·5	2970·5	72·5	4128·2
50·75	2022·40	61·75	2994·7	72·75	4156·7
51·	2042·82	62·	3019·0	73·	4185·3
51·25	2062·46	62·25	3043·4	73·25	4214·1
51·5	2082·88	62·5	3067·9	73·5	4242·9
51·75	2103·30	62·75	3092·5	73·75	4271·8
52·	2123·72	63·	3117·2	74·	4300·8
52·25	2144·14	63·25	3142·0	74·25	4329·9
52·5	2164·56	63·5	3166·9	74·5	4359·1
52·75	2184·98	63·75	3191·9	74·75	4388·4
53·	2206·18	64·	3216·9	75·	4417·8
53·25	2226·60	64·25	3242·1	75·25	4447·3
53·5	2247·81	64·5	3267·4	75·5	4476·9
53·75	2269·02	64·75	3292·8	75·75	4506·6
54·	2290·22	65·	3318·3	76·	4536·4
54·25	2311·43	65·25	3343·8	76·25	4566·3
54·5	2332·63	65·5	3369·5	76·5	4596·3
54·75	2353·84	65·75	3395·3	76·75	4626·4
55·	2375·83	66·	3421·2	77·	4656·6
55·25	2397·04	66·25	3447·1	77·25	4686·9
55·5	2419·03	66·5	3473·2	77·5	4717·3
55·75	2441·02	66·75	3499·3	77·75	4747·7
56·	2463·01	67·	3525·6	78·	4778·3
56·25	2485·01	67·25	3552·0	78·25	4809·0
56·5	2506·99	67·5	3578·4	78·5	4839·8
56·75	2528·98	67·75	3605·0	78·75	4870·7
57·	2551·76	68·	3631·6	79·	4901·6
57·25	2573·75	68·25	3658·4	79·25	4932·7
57·5	2596·53	68·5	3685·2	79·5	4963·9
57·75	2619·30	68·75	3712·2	79·75	4995·1
58·	2642·08	69·	3739·2	80·	5026·5
58·25	2664·86	69·25	3766·4	80·25	5058·0
58·5	2687·63	69·5	3793·6	80·5	5089·5
58·75	2710·41	69·75	3821·0	80·75	5121·2
59·	2733·97	70·	3848·4	81·	5153·0
59·25	2756·57	70·25	3875·9	81·25	5184·8
59·5	2780·31	70·5	3903·6	81·5	5216·8
59·75	2803·58	70·75	3931·3	81·75	5248·8
60·	2827·4	71·	3959·2	82·	5281·0

TABLE VII.—*continued.*

Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.	Diameter in Inches.	Area in Sq. Inches.
82·25	5313·2	88·25	6116·7	94·25	6976·7
82·5	5345·6	88·5	6151·4	94·5	7013·8
82·75	5378·0	88·75	6186·2	94·75	7050·9
83·	5410·6	89·	6221·1	95·	7088·2
83·25	5443·2	89·25	6256·1	95·25	7125·5
83·5	5476·0	89·5	6291·2	95·5	7163·0
83·75	5508·8	89·75	6326·4	95·75	7200·5
84·	5541·7	90·	6361·7	96·	7238·2
84·25	5574·8	90·25	6397·1	96·25	7275·9
84·5	5607·9	90·5	6432·6	96·5	7313·8
84·75	5641·1	90·75	6468·2	96·75	7351·7
85·	5674·5	91·	6503·8	97·	7389·8
85·25	5707·9	91·25	6539·6	97·25	7427·9
85·5	5741·4	91·5	6575·5	97·5	7466·2
85·75	5775·0	91·75	6611·5	97·75	7504·5
86·	5808·8	92·	6647·6	98·	7542·9
86·25	5842·6	92·25	6683·8	98·25	7581·5
86·5	5876·5	92·5	6720·0	98·5	7620·1
86·75	5910·5	92·75	6756·4	98·75	7658·8
87·	5944·6	93·	6792·9	99·	7697·7
87·25	5978·9	93·25	6829·4	99·25	7736·6
87·5	6013·2	93·5	6866·1	99·5	7775·5
87·75	6047·6	93·75	6902·9	99·75	7814·7
88·	6082·1	94·	6939·9	100·	7854·0

If the areas of larger cylinders are required, they will be found by the following RULE:—Multiply the square of the diameter in inches by the decimal ·7854, and the product will be the area in square inches; or, multiply half the circumference by half the diameter.

PART THIRD.

SECTION I.

DIRECTIONS FOR ASCERTAINING FROM THE DIAGRAM THE POWER EXERTED BY THE ENGINE.

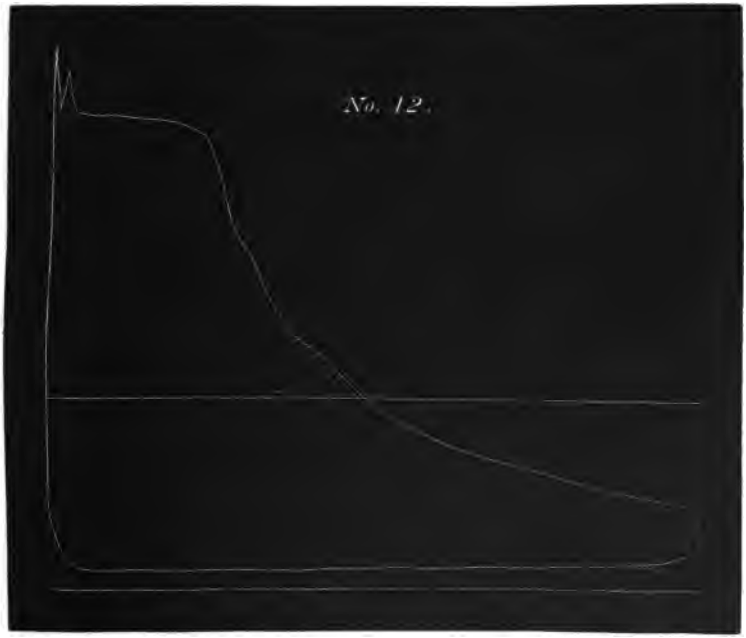
THE custom was introduced by Watt, and has since been generally followed in England, to designate the *size* of engines in measures of "horse-power." Watt ascertained by experiment that the power of London draught-horses, exerted with ordinary continuance, was to lift 33,000 lbs. one foot in one minute, and this is now employed, wherever English measurements are used, as the unit of measurement of the *actual* power of steam-engines.

When this measurement was introduced, steam was used only at the atmospheric pressure of 14·7 lbs. on the square inch, which is the unit of area commonly employed. Of this pressure, 4·7 lbs. were considered to be lost by imperfect condensation, and 3 lbs. by the friction of the engine, leaving 7 lbs. for effective pressure upon the piston, and the speed of piston employed was about 220 feet per minute. At the present day, pressures are employed varying from one to ten or twelve atmospheres, the former, however, being now rarely met with, and the speeds of piston range from 220 to 1000 feet per minute. Originally, the number of horse-powers defined at once the size and the power of an engine; but when a variety of pressures and speeds came to be employed, the same expression could no longer answer both of these purposes, and a distinction was introduced, and still prevails, between the "nominal" and the "actual" horse-power, the former being applied to the *size* of engines, irrespective of the pressure or speed employed, and the latter to the *power* which they exert. The term "nominal horse-power" has, moreover, acquired a variety of significations

in different localities, and it has become difficult to tell, in any case, precisely what is meant by it; but, fortunately, we shall have no occasion to make any further reference to it, as it is entirely a commercial expression.

The Indicator furnishes one of the data for ascertaining the actual power exerted by the steam-engine; namely, the mean or average pressure of steam during the stroke on each square inch of the piston, or, more accurately, the excess of pressure on the acting side of the piston to produce motion, over that on the opposite side to resist it.

It is of no consequence, in this respect, what the character of the diagram may be, whether most wasteful, like the one



shown in fig. 13, page 88, or most economical, like fig. 12; for the purpose of ascertaining the power exerted, we have merely to measure its included area, and so get the mean pressure on a square inch during the stroke, which this area represents. This pressure being multiplied into the number of

square inches, we have the total number of pounds of force exerted. This force is acting through the distance travelled by the piston. We multiply it by the distance in feet through

which the piston travels in one minute, and the product is the number of foot-pounds of force exerted in one minute. This, divided by 33,000, gives the number of horse-powers. It is to

be observed that in this calculation force and distance are treated as convertible. However extremely unequal, as in fig. 8, page 89, the pressures may be at different points of the stroke, these are all reduced to an average pressure, which is conceived to be uniformly exerted through the stroke. Then, finally, all the power exerted in a minute is conceived as a certain number of pounds of force exerted through one foot.

The above calculation gives what is called 'the indicated power' of the engine. Concerning this we are to observe, first, that it is not the gross power exerted. The included area of the diagram represents only the difference between the opposing forces which act to produce and to resist the motion of the piston. The force of the steam must in all cases be first applied to overcome what is called the back pressure. In a non-condensing engine this must be at least the pressure of the atmosphere. It is always, in fact, more than this by the amount of force that is required to expel the exhaust steam through the port, passages, and pipe against the resistance of the atmosphere. Sometimes the excess of back pressure above that of the atmosphere is scarcely perceptible, as in Diagram No. 11. In a badly constructed engine, on the other hand, the force required for this purpose may be very great, as in diagram No. 13, which is almost too bad in this respect to be credited, but the original of which is preserved by the writer among mementos of the Exhibition of 1862. The usefulness of the Indicator in revealing defects of this nature can hardly be estimated. It has been seen that it showed in the locomotive engines of the London and South-Western Railway, as they were running twelve years ago, a back pressure of 10 lbs. above that of the atmosphere. The office of the condenser and air-pump is to remove the back pressure or resistance of the atmosphere from the piston of the engine to the piston or plunger of the air-pump; by which means indeed it is, to the extent of the vacuum obtained, got rid of altogether; since the atmosphere exerts there the same force to produce motion in one direction that it does to oppose it in the contrary one. But in all cases it is only the net power exerted, after deducting that necessary to overcome the back pressure, that is represented in the included area of the diagram.

But, second, we are to observe, that neither does the diagram represent the effective power given off by the engine.

Friction.—To ascertain the effective power of an engine, it is necessary to deduct from the power represented by the diagram that which is expended in overcoming the friction of the engine

itself. This is properly divisible into two parts: first, the power required to drive the engine alone, or idle; and, second, that required to overcome the increased friction, if any, in the guides and bearings when it is running under the weight of its load.

The first of these we can generally ascertain, in the case of stationary engines, by taking what are called frictional diagrams. Diagram No. 8, is a diagram supposed to represent the power exerted in driving an engine, and an idle line of shafting. Sometimes it is, as it was in this case, impos-

sible to relieve the engine of *all* its load. The expansion line crosses the line of counter-pressure at B; the net power is the difference of the two areas. This may as well be explained here. The expansion of the steam takes place in the closed cylinder, and goes on without reference to any external con-

ditions of pressure whatever, and in this case we see the pressure falling to a point considerably below that of the atmosphere. The back pressure on the non-acting side of the piston can never be got rid of. The return line of the diagram shows this back pressure during the return stroke. Assuming it to have been the same during the forward stroke, while this expansion line was being drawn, it is evident that during six-tenths of the stroke it was greater than the forward pressure, and so the net force, or difference of the two opposing forces, was exerted through this period to stop the engine. Now the force exerted during the first four-tenths of the stroke in the forward direction, had first to be applied to neutralise this retarding force, and it was only the excess which was available to produce motion. This will appear still more clearly by-and-by, when we come to treat of the real diagram.

In taking the frictional diagram, extreme care should be observed that, at the moment when it is taken, the speed of the engine is not being in the least degree accelerated or retarded. We want to learn with certainty the force that is required to maintain uniform motion. But if the motion is being accelerated, or if, on the other hand, it is not being maintained, obviously the diagram will be larger or smaller than the true frictional diagram, and may be so in almost any degree.

If the Indicator is left to run for some time, and the diagram steadily repeats itself, then its truth in this respect is demonstrated, as it cannot be so conclusively in any other way. In many engines the diagram will not repeat itself precisely. Then no single figure can represent the power being exerted. It is necessary in such cases that a sufficient number of successive figures be drawn, when the mean of all these will be the true diagram.

Where the valves work under pressure, the full pressure must be admitted to the chest when the frictional diagram is taken, as in diagram No. 8. It is very convenient to throttle the steam, and admit only the trifling pressure necessary to drive the engine when the steam is following the piston as far as the valve-movements will permit; but this is admissible only when equilibrium valves are employed. The pressure required to drive an engine alone depends greatly on its construction and condition. If well made, and in good running order, one pound on the square inch will often be sufficient,

irrespective of that required to drive the air-pump. In great engines this should be sufficient to drive the air-pump also. The speed at which the engine is run seems to make no appreciable difference in this respect, and, what appears more singular, the power required is not observed to increase with increase in the size of the bearings. Probably the greater extent of the surface compensates, by the superior lubrication, for its increased velocity. A slight derangement, or want of lubrication, causes a marked increase in the frictional diagram. In no other case is the exquisite exactness of the representation made by the Indicator seen so well as here.

The increase in the friction of the guides and bearings when the engine is running under the weight of its load, depends on the same conditions of construction, &c. which affect the resistance that the engine meets when running alone. Little is really known about it.

There is no doubt that, other things being equal, the friction will vary directly as the pressure; but often the pressure in the bearings is actually less when the engine is driving its load than when it is running idle. The strain of a belt, or the force exerted on the teeth of gears, may tend directly to lift the fly-wheel, transferring a portion of the weight to the bearings of a smaller shaft; a point not undeserving the attention of engineers seeking to attain the best results. The reader who shall make himself acquainted with Part V. of this treatise will see the conditions under which the friction on the crank-pin and main bearing is not increased at all by loading the engine. Experiments with the friction-brake have shown no appreciable difference between the losses of power in friction when very small and when very heavy loads were being driven by the same engine. There is nothing for confounding superficial reasoning like an experiment.

We will now describe the mode of ascertaining from the diagram the mean pressures on the opposite sides of the piston during the stroke, in condensing and in non-condensing engines.

Divisions of the diagram.—For this purpose divide the diagram, first of all, into ten equal parts, by lines drawn perpendicular to the atmospheric line. Sometimes, as in diagram No. 8, it will be necessary to subdivide some of these divisions, and perhaps to divide again the subdivisions so obtained, and very nice

observers may divide the whole diagram into twenty or more parts, but the basis is always the division of the length of the diagram into ten exactly equal parts, and commonly this division is sufficient. A convenient instrument for facilitating this operation, saving time, and insuring accuracy, is furnished with these Indicators. It consists of a parallel ruler of eleven bars of thin steel, and a small square. A line perpendicular to the atmospheric line is first drawn by the square at one end of the diagram, when, the outer edge of bar No. 1 being brought to this line, and the inner edge of bar No. 11 to the opposite end of the diagram, the dividing lines are drawn with a sharp-pointed pencil, or, on the metallic paper, with a common pin. If twenty divisions are desired, the intermediate lines for this purpose will also be readily drawn by means of this instrument, points being first marked in the middle of the outer divisions. It is an excellent practice to divide the diagram into equal divisions also, by lines drawn parallel with the atmospheric line, each division representing a certain number of pounds pressure, generally five or ten, and the lines being numbered on the margin according to the scale of the Indicator, and continued upward to the boiler pressure. By this means the engineer is able to observe more accurately the general nature of the diagram. The same instrument may be employed also for this purpose.

On diagrams from condensing engines the line of perfect vacuum should also be drawn at the bottom. The line of perfect vacuum varies in its distance from the atmospheric line, or, more correctly, the latter varies in its distance from the former, according to the pressure of the atmosphere, as shown by the barometer, from 13·75 lbs. on the square inch when the mercury stands at 28 inches, to 15 lbs. when it stands at 30·54 inches (*vide* Table No. III. page 58); and it should be drawn according to the fact, if this can be ascertained. The engineer should always have a good aneroid in his pocket. The pressure of the atmosphere is usually reckoned at 15 lbs., which is too high, being correct only when the barometer stands at 30·54 inches—a most unusual occurrence; but the error is unimportant, and it is very convenient to avoid the use of a fraction, and to say that 30 lbs., 45 lbs., 60 lbs., and so on represent 2, 3, 4, 5, 6 atmospheres of pressure.

The principal object of knowing the exact pressure of the

atmosphere is to ascertain the duty performed by the condenser and air-pump. The temperature of the discharge being known, the pressure of vapour inseparable from that temperature is also known (*vide* Table No. III. page 58), and this being deducted from the actual pressure of the atmosphere, the remainder is the vacuum in which the water would boil. The power of the air-pump is shown in the closeness with which the vacuum approaches this point.

Measurement of the diagram.—The diagram having been accurately divided, in the manner above explained, we are now ready to measure the mean pressure shown in each one of the equal divisions; and these being added together, and their sum divided by the number of divisions, we shall have the average pressure during the stroke. To measure the pressures with exactness requires a good eye and some practice. If the upper and lower boundaries of a division are straight lines, then, whatever angles these may form with the sides or dividing lines, the mean pressure will be accurately measured between their middle points. But often the upper or lower line is curved, or broken, or irregular, and then it is necessary first to draw or imagine a straight line, which will include the same area that is included by the line on the diagram, and then we can take the middle point of this line to measure from. When a waving line is drawn by the Indicator we can draw in by hand the real line, which is the mean of the oscillations; but the better way is to follow the oscillations themselves, and measure the area of the very figure the Indicator has described. It is, however, sometimes necessary, as in fig. 29, page 107, to draw in the line representing the mean of a serrated line at the commencement of the stroke, and to measure from a point on that line.

Diagram No. 8 presents a reasonable number of difficulties and will afford a good lesson in measurement. It is necessary to subdivide the first two divisions, and to divide the first one of these subdivisions again. After this has been done the first operation is to mark, wherever it is necessary, in the middle of each division, the points between which to measure. These are seen in the figure, and each may be considered as marking the position of the horizontal line, which, if drawn between the nearest perpendiculars, would include the same area that is included by the line of the diagram.

The measurement of the first four-tenths of this diagram is as follows :—

Divisions.	Average pressure during each division.
·25	43·5
·25	30·5
·5	74·0 ÷ 2 = 37·
·5	.. .. 24·5
1· ..	.. .. 61·5 ÷ 2 = 30·75
·5	.. .. 13·
·5	.. .. 9·5
2· ..	.. .. 22·5 ÷ 2 = 11·25
3· ..	.. .. 4·75
4· ..	.. .. 1·75
Forward pressure, averaged for the stroke ..	4·850 lbs.

This is all the forward pressure that we have got; as, just at the end of the fourth division, the expansion curve crosses the line of counter-pressure. If we were to divide the sum of the pressures by four, the number of equal divisions through which the forward pressure continues, we would have a mean pressure of 12·125 lbs. during that portion of the stroke. This, however, would be of no use to us: we want to know what this pressure would average, supposing it to be extended over the entire stroke. We get this by dividing the sum of the mean pressures by 10, the whole number of equal divisions, and the result is 4·85 lbs. The correctness of this method will be apparent if we take the mean pressures for each division, and find what would be the equivalent pressure continued through the stroke, and add these together.

These would be as follows :—1·0875
·7625
1·225
·650
·475
·475
·175

4·850 lbs. as before.

The mean pressure to be deducted from this is ascertained in a similar manner, and is as follows:—

5th division	1·5
6th "	3·
7th "	4·25
8th "	5·
9th "	5·
10th "	4·25
	2·300 lbs.

leaving the apparent mean pressure 2·55 lbs. on the square inch. We say the *apparent* mean pressure, because this diagram did not tell the truth, and one reason for introducing it here has been to illustrate an error, and show the extreme care that must be taken if we wish to learn the truth, and not to be misled by the diagram.

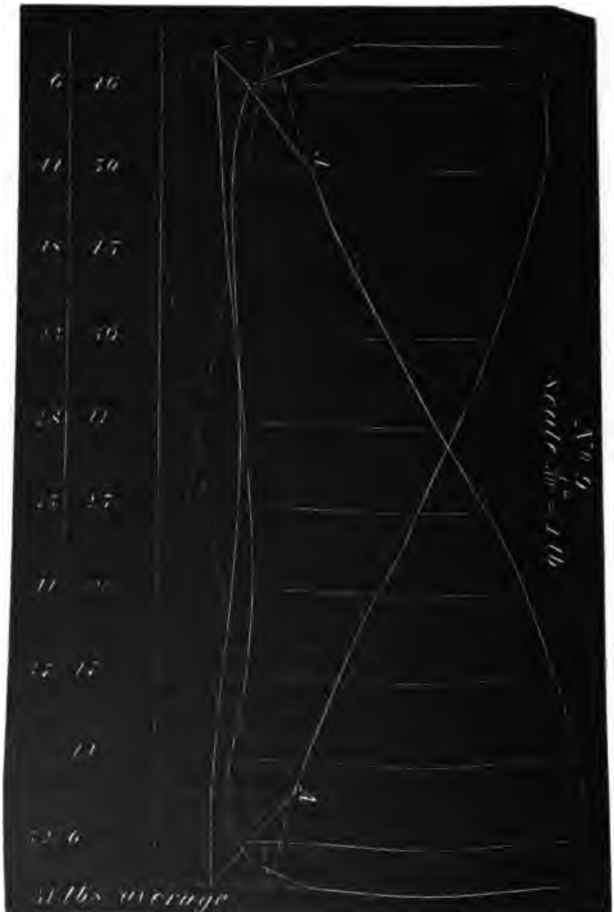
How few, after all, are the cases in which all the care is taken that ought to be to ensure accuracy! Engineers of experience know better than to conclude anything with certainty about an engine from diagrams that have not been taken by themselves.

In this case the pencil first touched the paper at A, but was held so hard against it that on the next stroke it did not come down to the same point, and so the diagram was good for nothing, and by the merest accident showed its own worthlessness. This is a common source of error. The motion may be entirely frictionless, and no lost motion may be possible, and all adjustments may be correct; and yet a careless operator may manage in this way to get a false diagram. There is especial need to guard against the friction of the pencil when indicating on a large scale. It is an excellent practice, after having taken a diagram, to withdraw the pencil a little from the paper, and observe its movements: it will at once be seen whether it retraces the same line or not: nearly right is wrong.

The method above described is the only correct way of measuring the mean pressure. Whatever the form of diagram and it is very curious sometimes, this method will give the mean pressure precisely.

A method sometimes employed is shown on these diagrams

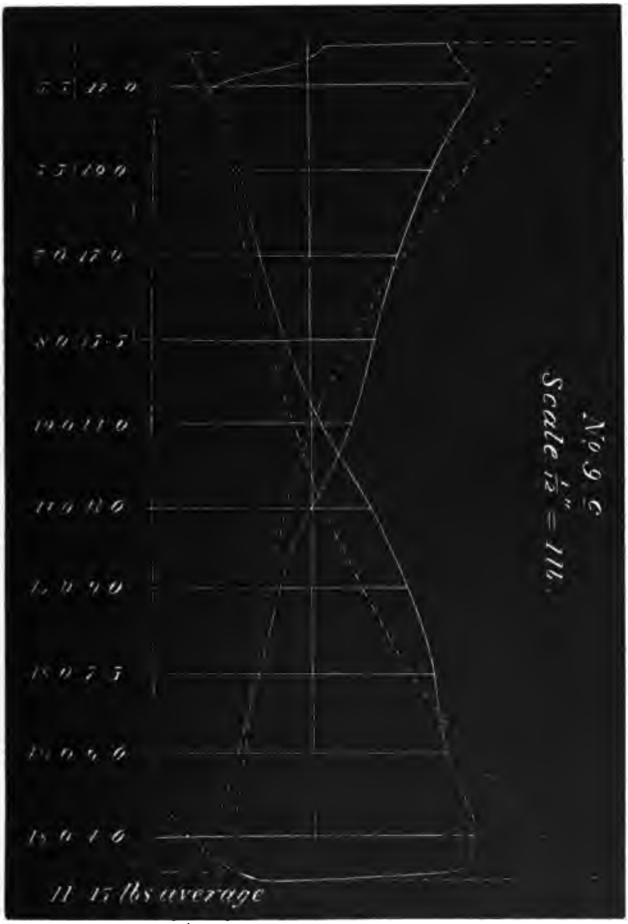
No. 9 and No. 9 *c*, from the compound cylinder engines of the steamship *Egypt*. They are shown as they came to the writer. Having been divided into ten equal parts, perpendiculars were drawn through the middle of each division, and the pressure was measured on each of these lines. This method gives only



an approximation to the truth, more or less close, as the diagram is more or less regular in its form. Any method whatever would give a correct result if the diagram were a parallelogram.

The test of the excellence of a method is found in the degree of its adaptation to difficult and complicated cases.

In measuring diagrams from condensing engines, it is usual to measure the portion above and that below the atmospheric line separately. This is conducive to accuracy, as generally



there will remain but one side of the area being measured that is irregular in outline. Sometimes this method also will give the average vacuum realised, and so tell us in what degree the resistance of the atmosphere is removed from the non-acting

side of the piston by those parts of the engine whose function this is; but often it will be the case, as in diagram No. 12, p. 85, that the expansion curve crosses the atmospheric line, and perhaps, as there shown, at an early point of the stroke. Then it is evident that the mean pressure represented by the included area below the atmospheric line is less than the vacuum attained in the cylinder.

Calculation of the power.—For this purpose we are now done with the diagram. It has shown us the pressure on each one square inch of the piston, at each point in its stroke, and we have found the method of determining what this pressure, however unequal it is at different points, would be if it were averaged and made uniform during the stroke. We now proceed to use this quantity, called *the mean pressure*, in calculating the power of the engine. We assume it to be correct; whether or not it is so, depends upon the care with which the diagram has been taken and the reduction been made. The latter is open to revision, but not so the former. Sometimes, though rarely, the diagram may give to the practised eye an indication of its own inaccuracy, but even then affords no guide to the correction of it.

We have now to find the area of the piston in square inches. Knowing its diameter exactly, we take the area from the Table, page 80; from this is to be deducted one half the area of the rod, the remainder is the average area of the two faces. We multiply this by the mean pressure on the square inch, and the product is the total constant force under which the piston is moving, or which is acting through the distance travelled by the piston. This being multiplied into the distance, in feet, through which the piston travels, or through which the force acts, in one minute, gives the foot-pounds of power developed, or of work done, in that time, and this sum, divided by 33,000, gives the number of horse-powers exerted.

It is interesting to consider the variety of the conditions out of which this result is derived. We have, first, every variation of pressure, from the highest to the lowest; and second, in combination with this, every different speed of piston, from infinitely slow up to the velocity of the crank. The latter variation is not regarded. Forces different in amount are separated forces. The diagram tells us what each separated force was, and through what distance it acted, and this is all we require for the computation of power. Each force being multiplied

into the distance through which it acted, and the product divided by the length of the cylinder in units of such distance, the sum of all is the pounds of force acting through the length of the stroke. That some forces were exerted for a longer time than others in acting through an equal distance is nothing. Static forces, though exerted for ever, have no dynamical value. Force acquires this value only as it acts through distance.

The better mode of computing power is, first, to obtain for any engine a constant, which, being multiplied into the mean pressure, will give the number of horse-powers. This constant is the number of horse-powers which would be exerted by 1 lb. of mean pressure. It is found by multiplying together the area of the piston in square inches and the feet travelled by it per minute, and dividing the product by 33,000.

In illustration of the above rules we will compute the horse-powers exerted in the following cases:—

Example 1.—What was the indicated power exerted by the pair of engines, from one of which diagram No. 1, page 12, was taken, it being assumed that the other three diagrams from the two cylinders would have been similar to this one?

The diameter of each piston was 95 inches, the stroke 10 feet, and the revolutions 15 per minute, and the diameter of the rod 8 inches. The area of the piston, less one-half that of the rod, was 7063·07 square inches, and the distance travelled by it in feet per minute was 300. The constant for these two cylinders was, therefore, $7063 \cdot 07 \times 300 \times 2 \div 33{,}000 = 128 \cdot 42$ horse-powers exerted by each 1 lb. of mean pressure.

The mean pressure is figured on the diagram, and the power exerted was as follows:—

Above the atmospheric line, $9 \cdot 82$ lbs. $\times 128 \cdot 42 =$	1261·0844
Below the atmospheric line, $11 \cdot 46$ lbs. $\times 128 \cdot 42 =$	1471·6932
Total indicated horse-powers	2,732·7776

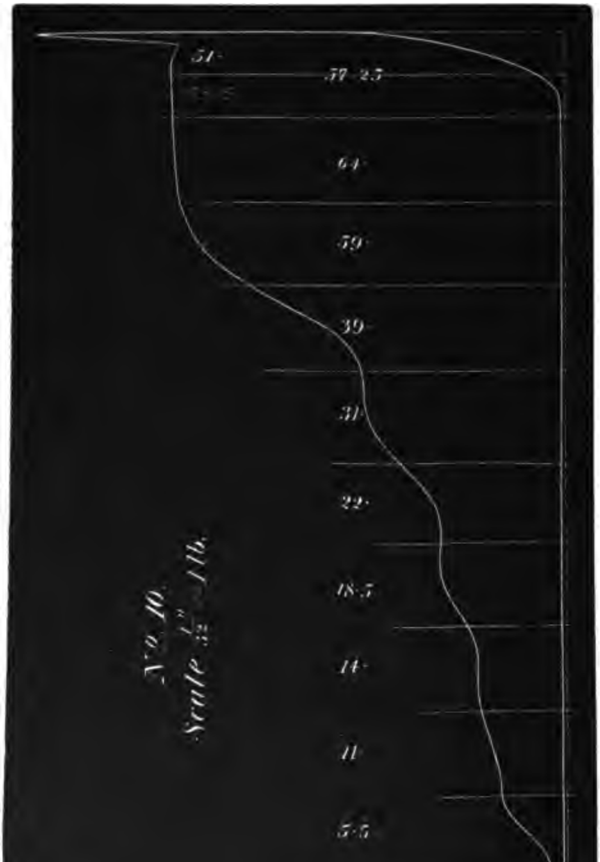
of which it will be observed that the vacuum furnished considerably the larger portion.

Example 2.—What was the indicated power exerted by the engine from which diagram No. 10 was taken, assuming the opposite diagram to have been the same?

The piston is 12 inches in diameter, stroke 20 inches, revolu-

tions 160 per minute, and diameter of the rod 2 inches. The area of the piston, less one-half that of the rod, is 111·53 inches, and the distance travelled by it in feet per minute is 533·33 feet. The constant for this engine is, therefore,

$$111 \cdot 53 \times 533 \cdot 33 \div 33{,}000 = 1 \cdot 8025.$$



The mean pressure is figured on the diagram, 32·125 lbs., and the power exerted was,

$$32 \cdot 125 \times 1 \cdot 8025 = 57 \cdot 9 \text{ H.P}$$

In this simple manner the power exerted by an engine may be ascertained under every variety of circumstances, and also that required by every kind of machinery. Measuring the power required by a single machine among many running in a manufactory requires great care, but can be done with certainty, even to a small fraction of a horse-power. It is necessary that everything else should be known to be in the same condition during the whole experiment. The proper time is after running for several hours, and directly after stopping, when everything is in the best running condition: at noon-time, perhaps. Then, first indicate for the shafting alone, afterwards add the machine, the requirement of which is to be measured, and indicate the engine after this has been running for a few minutes, and, finally, after that has been stopped, indicate for the shafting again. In each case the pencil should run over the figure several times, and afterwards be observed if it follows it exactly when removed a little from the paper. The first and third diagrams should be identical, and the excess of the second one is the power required by the machinery tested. It is wonderful how much guess-work there is, for which exact knowledge might thus easily be substituted. Care should be observed that all the diagrams are taken at the same speed of the engine.

In the case of any engine, condensing or non-condensing, the power required to overcome the counter-pressure is found by multiplying such counter-pressure in pounds on the square inch measured from the line of perfect vacuum, by the constant for that engine.

Always, after a diagram has been taken, the careful observer will take pains to determine if it is the true representation of the power being exerted, by trying if the pencil will repeat it, both when in contact and when not in contact with the paper. Often the diagram will not be repeated exactly. Whenever this is the case, the pencil must be allowed to run over the paper a sufficient number of times, and the average of all the figures must be taken as the true one.

SECTION II.

TO MEASURE FROM THE DIAGRAM THE QUANTITY OF STEAM CONSUMED.

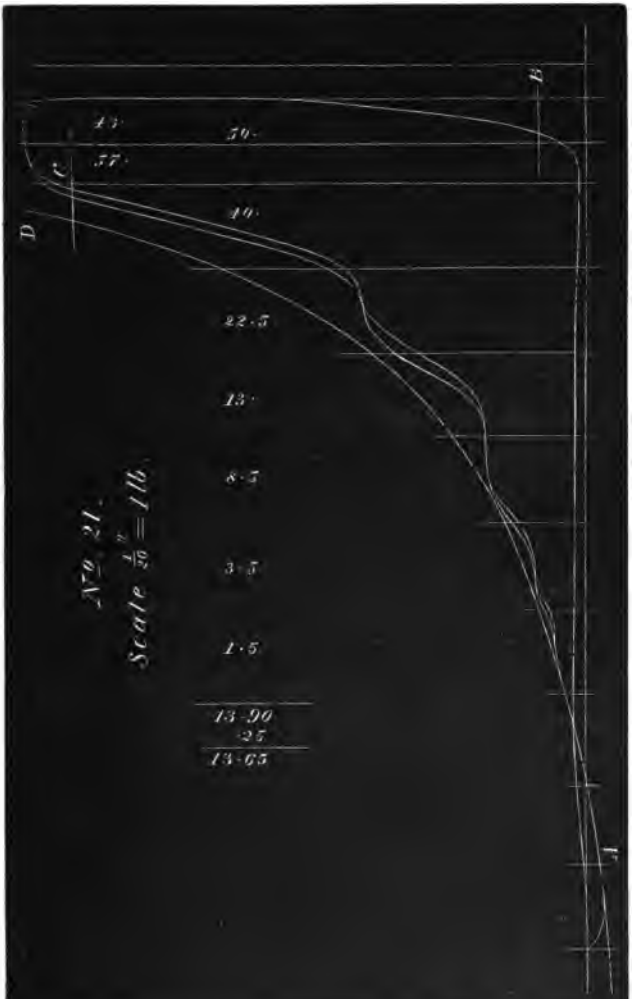
As the mean pressure during the stroke measures the work done, so the pressure at the end of the stroke measures the steam consumed in doing it. By means of the Indicator we are able to ascertain the weight of the water existing in the form of steam in the cylinder at every point in the stroke, not absolutely—since we do not know absolutely the weight of steam at different temperatures—but, without doubt, very nearly. This, when measured just before the opening of the exhaust, is the weight of water accounted for by the Indicator. From a variety of causes, the weight so accounted for can never be the full weight required to supply the boiler. These causes will be briefly considered at the end of this section. At present we will confine our attention to the method of ascertaining the weight of the steam, of which the Indicator shows the pressure.

For this purpose we have first to determine the volume of the steam, or the capacity of the chamber which it fills. This is the product of the cross-section area of the cylinder into the length of the stroke, measured up to the point at which the pressure of the steam is taken; to which is added the waste room in the clearance and passages at one end of the cylinder. This capacity, generally obtained in cubic inches, is to be reduced to cubic feet. The pressure of the steam, reckoned from the line of perfect vacuum, which is always to be drawn or imagined as already directed, is then to be taken at that point of the stroke to which the volume has been measured, and which should be the latest point at which it is certain that the exhaust has not commenced. The reason for taking this point will appear by-and-by.

The weight of one cubic foot of steam at this pressure will be found in the Table. This being multiplied by the volume, the product is the weight of the steam contained in the cylinder.

From this there is to be deducted the weight of the steam saved, by being confined in the cylinder by the closing of the

exhaust. This is ascertained in the same manner, the pressure being taken at a point after it is certain that the exhaust is closed, and the weight of one cubic foot at the pressure existing



at that point being multiplied into the volume, in cubic feet, of the remainder of the return stroke, as measured on the diagram and the waste room. This having been deducted, the remainder

is the weight of steam consumed per stroke, as accounted for by the Indicator; from which may be learned the quantity so accounted for per minute, or hour, per horse-power exerted, and pounds of coal burned, and the proportion which it forms of the water supplied to the boiler.

Example.—What was the weight of steam consumed in the cylinder from which diagram No. 21, page 103, was taken; the cross-section area being, as the mean of both ends, 198·6 square inches, the stroke 30 inches, and the waste room 248 cubic inches, and the engine making 125 revolutions per minute?

Let the pressure be measured at ·9, or 27 inches of the stroke. This point is marked A on the diagram. The volume of steam at this point is,

$$\frac{(198 \cdot 6 \times 27) + 248}{1728} = 3 \cdot 2466 \text{ cubic feet.}$$

The pressure is 13·5 lbs. The weight of 1 cubic foot of steam at this pressure is ·035 of a pound, and 3·2466 × ·035 = ·11363 of a pound. From this there is is to be deducted the steam saved, as follows:—

We take, on the diagram, a point say at $\frac{3}{16}''$ from the end of the return stroke, and through it draw the horizontal line B. This distance represents 1·28″ on the return stroke, it bearing the same proportion to 4·4″, the total length of the diagram, that 1·28 bears to 30.

$$\frac{(198 \cdot 6 \times 1 \cdot 28) + 248}{1728} = \cdot 29 \text{ of a cubic foot.}$$

The pressure at this point is just 20 lbs. The weight of a cubic foot of steam at this pressure is ·0511 of a pound. This multiplied by ·29 = ·01482. The weight of steam accounted for by the Indicator per stroke was, therefore,

$$\cdot 11363 - \cdot 01482 = \cdot 09881 \text{ of a pound,}$$

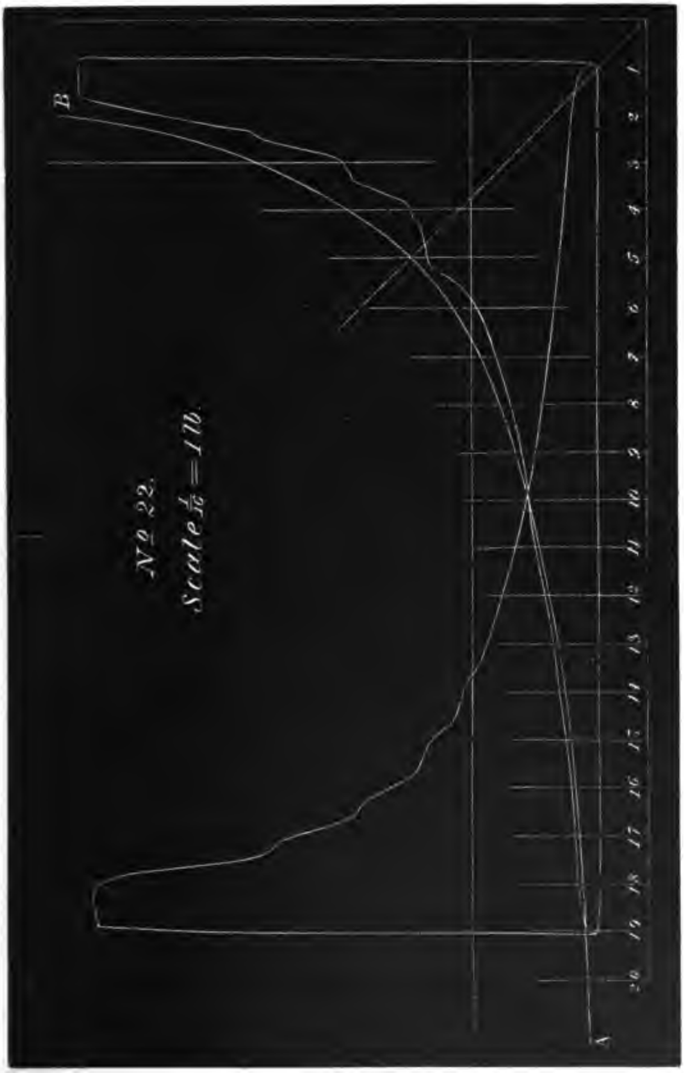
and the quantity per hour was ·09881 × 250 × 60 = 1482·15 lbs. How much was this per indicated horse-power? The constant for that engine was 198·6 × 625 ÷ 33,000 = 3·76. The mean pressure is figured on the diagram.

$$13 \cdot 65 \times 3 \cdot 76 = 51 \cdot 32 \text{ indicated horse-powers.}$$

And the water consumed per horse-power per hour, so far as it was shown by the indicator, was,

$$1482 \cdot 15 \div 51 \cdot 32 = 28 \cdot 8 \text{ lbs.}$$

In contrast with this we will compute, in the same manner, the weight of steam consumed in the same engine when diagram



No. 29 was taken. The comparison of these diagrams will afford a good practical lesson in expansion.

We employ the constants for volume already obtained. The pressure at ·9 of the stroke was 26·7 lbs. The weight of one cubic foot of steam at this pressure is ·067 of a pound.

$$3·2466 \times ·067 = ·2175 \text{ of a pound.}$$

The steam saved is as follows:—

Pressure at 28·72 inches of the return stroke 20·2 lbs. The weight of 1 cubic foot of steam at this pressure is ·0513 lbs. $·29 \times ·0513 = ·0149$; and $·2175 - ·0149 = ·2026$ of a pound. $·2026 \times 250 \times 60 = 3040·2$ lbs. of water consumed per hour.

How much was this per indicated horse-power? The mean pressure is figured on the diagram, $38·07 \times 3·76 = 143·143$ H.P. $3040·2 \div 143·143 = 21·24$ lbs. of water per horse-power per hour.

In the former case the steam was expanded 7 times, and in this one only about 3½ times; and yet in the former 28·8 lbs. of water were consumed per hour to supply one indicated horse-power, against only 21·24 lbs. in the latter, or about 4 to 3.

There were two principal causes operating to turn the gain which ought to be derived from the greater expansion into this great loss. The consideration of one of these causes, namely, the greater proportionate loss of heat suffered by the smaller body of steam in restoring the heat of the cylinder, belongs to a future chapter; but the principal cause was the following:— The water consumed is divided among the indicated horse-powers. Now the power required to overcome the back pressure, power thrown away, was the same in each of these cases, namely, $15·5 \times 3·76 = 58·28$ horse-powers. This, in the first case, was nearly 7 horse-powers more than the indicated power, being 53·18 per cent. of the whole power exerted; while in the latter it was only 29 per cent. of the whole.

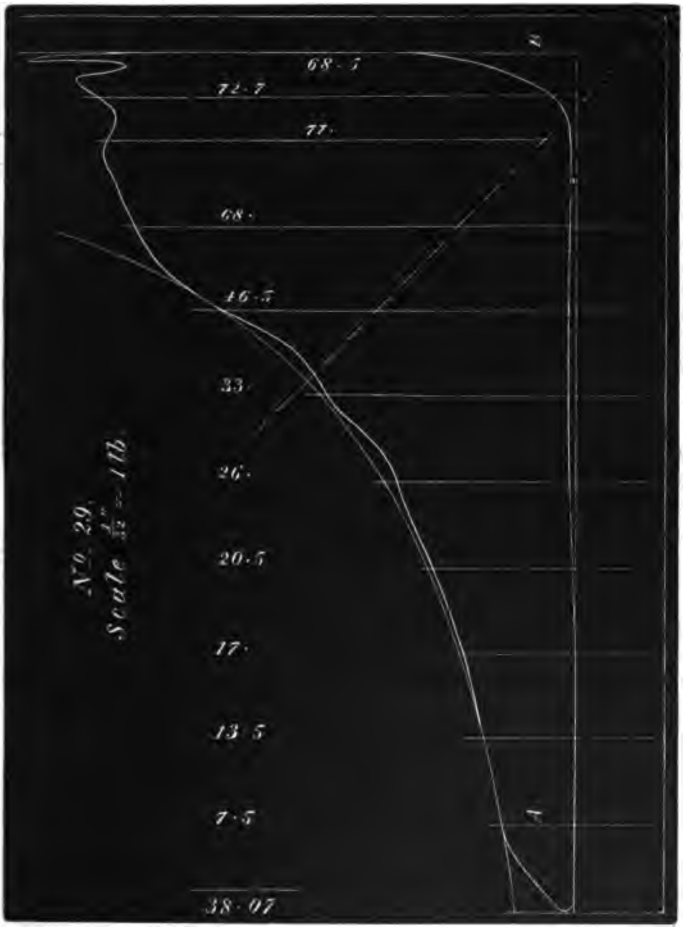
We can ascertain, also, how much loss has resulted from the greater proportionate condensation of the entering steam in the former of these cases. If we divide, in each case, the steam consumed by the *gross* power exerted, the results will be as follows:—

$1482·15 \div 109·6 = 13·5$ lbs. per horse-power per hour.
$3040·2 \div 201·4 = 15·0$ lbs. " "

Now the ratio of gain from a 3·5-fold expansion is 2·253, and that from a 7-fold expansion is 2·946. So if the advantage from expansion had been realised, as well in the case of the earlier

cut-off as it was in that of the later one, the consumption would have been in that case only 11·47 lbs. instead of 13·5 lbs. per horse-power per hour (2·253 × 15·) ÷ 2·946 = 11·47.

The above comparisons illustrate two of the principal causes



which operate to limit the gain to be derived from expansion. The first of these is in a great degree removed by condensing, which indeed has this for its sole object. The apparent gain effected in this way is not all real, however, since condensing increases the loss arising from the cooling of the cylinder.

When the water accounted for by the Indicator, ascertained in the manner above explained, is compared with that required to supply the boiler, a deficiency will always be found, and sometimes a great one. The principal causes of this are the following:—

First.—A certain amount of water always disappears from a boiler in ways which cannot be accounted for. If a boiler is shut perfectly tight, without visible outlet for any steam whatever, and a steam pressure is maintained in it, the water will gradually subside. When experiments are to be conducted, the rate of this disappearance from the boilers, under the pressure to be employed, ought to be ascertained.

Second.—Unless the steam is superheated, more or less water is carried over to the engine mechanically. This is especially the case with boilers which show a *great evaporative duty.*

Third.—As soon as the steam leaves the boiler it begins to be condensed. It can receive no more heat from any source, but it must impart heat to everything, and supply all loss from radiation.

Fourth.—A certain amount of condensation is produced by the conversion, during the expansion, of heat into mechanical work, of which we shall speak by-and-by.

Fifth.—A portion of the steam is always condensed as it enters the cylinder from coming in contact with the surfaces which have just been cooled by being exposed to the colder vapour of the exhaust, and especially by the evaporation at the same time of moisture from them, abstracting the heat necessary to supply to such moisture the heat of vaporisation. This also will be more fully considered in its appropriate place.

SECTION III.

TO MEASURE FROM THE DIAGRAM THE QUANTITY OF HEAT EXPENDED.

THE Indicator enables us also to ascertain the quantity of heat, measured in thermal units, contained in the steam; from which the quantity converted into mechanical work being deducted, the rest is thrown away. For this purpose we have to obtain the volume and pressure of the steam at a point as soon as possible after we know that the admission port has been closed. The heat contained in the steam at this point is not all that has entered the cylinder—an unknown quantity has gone to restore the temperature which the surfaces have just lost—but it is all about which we will now concern ourselves.

If the material of which the cylinder is constructed were a perfect non-conductor, then we might expect that, the heat converted into mechanical work during the expansion being deducted from the quantity contained in the steam at the point of cut-off, the balance should be the quantity contained in the steam at the point of release. Iron being, however, a pretty good conductor, it might be concluded that it would carry off some of this heat, so that at the end of the expansion there would not be so much in the steam. But the fact is that there is always more, and often a good deal more, than the above calculation would call for; that the more heat there is by any means imparted to the steam, the less it contains at the point of release; that the more it loses during the expansion, the more economically the engine works; and that no means have been found to bring the quantity down to that which the above computation requires, and it is demonstrable, as we shall see pretty soon, that none ever will be.

Let us refer again to diagram No. 21, page 103. The line C crosses the largest expansion line at 3·4" of the stroke. At this point the volume of the steam is ·5354 of a cubic foot, its pressure is 69 lbs., having fallen 4 lbs. during the closing of the port, and its weight is ·16598 × ·5354 = ·088866 of 1 lb. The number of thermal units that it contains is 1205·9744 × ·088866

= 107·17, which are equal to 82,735 foot-pounds of mechanical work.

The *total* work—for we must include the work done in over-

coming the back pressure—performed during the expansion, from this point to ·9 of the stroke, was equal to a mean pressure of

20·7 lbs. on the square inch through the stroke, or to 10,277·5 foot-pounds. There had already been exerted a force equal to 8·1 lbs. on the square inch through the stroke, or to 4,021·65 foot-pounds of power, before the piston had arrived at the point C. This was work done on the change of state from water to steam in the boiler, as will be explained presently, and it does not come into this calculation.

If, therefore, no heat were added to the above quantity during the expansion, then the quantity that we ought to find present at the point A would be 82,735 − 10,277·58 ÷ 772 = 93·85 thermal units. How many does it, in fact, contain at that point?

In the last section we found the weight of the steam at this point to be ·11363 of a pound. This weight we now observe to have been increased during the expansion by ·11363 − ·08886 = ·02477, or about 28 per cent. The pressure we found to be 13·5 lbs so that the heat contained in the steam is

$$1177 \cdot 294 \times \cdot 11363 = 133 \cdot 776 \text{ thermal units.}$$

Instead, then, of the quantity of heat contained in the steam having diminished during the expansion, when 13·32 thermal units were converted into mechanical work, it has increased by 133·776 − 107·17 = 26·606 thermal units, or very nearly 25 per cent.; a loss which, when known, is calculated to set an engineer upon inquiry as to what it is caused by, and how it is to be prevented.*

In this simple manner this interesting and instructive analysis can be applied to any diagram. The inquiry into the causes of these apparent anomalies will be made in connection with another branch of our subject.

* It will be seen by-and-by that there is an inaccuracy here, since at the point C 5·4 thermal units had been lost by the steam, having been converted into mechanical work. No error is involved, however, the thermal units as found in the Table being given at *both* ends of the expansion curve.

SECTION IV.

OF THE REAL DIAGRAM, AND HOW TO CONSTRUCT IT.

THE included area of the diagram is commonly supposed to represent the pressure on the piston at each point of the stroke. A moment's reflection will, however, show that it does not. What we, for convenience, call the upper and lower lines of the diagram, have, in fact, no relation to each other. To get a correct idea of the nature of the diagram, we must disabuse our minds of the confused notions which result from this inexact use of language. There is in reality no lower line ever drawn by the Indicator. The real lower line of the diagram is always the line of perfect vacuum. During a revolution the Indicator draws from opposite sides of the piston the upper lines of four separate diagrams, and the two which appear together as parts of the same outline are the ones which do not belong together, having no relation to each other whatever.

For example, on the forward stroke a line is drawn by the Indicator, showing at every point the height at which the pencil is raised by the pressure on that side of the piston upon which the steam is admitted. Beneath this line, at the proper distance, let the line of perfect vacuum be drawn, and the extremities of the two connected by lines perpendicular to the latter. We have now a correct and complete diagram of the pressure on that side of the piston during that stroke.

To illustrate this, we will take the diagram drawn with the heavy line in fig. 19, in the Appendix. The following figure, No. 30, represents the pressure on the acting side of the piston during the stroke when the upper line of that diagram was drawn.

The lower line of that diagram was commenced after the upper one was finished, and is, in truth, the upper line of another diagram of which also the line of perfect vacuum is the lower line, and which represents the pressure on the same side of the piston during the next stroke. The figure,

No. 31, page 114, drawn in the manner above directed, represents this second diagram.

We say that this last diagram represents the pressure exerted by the steam to oppose the return of the piston; in fact, the piston is always between opposing forces. It moves in obedi

ence to the stronger one. The weaker we distinguish as the opposing force, because it resists the motion which nevertheless takes place. At every point in the motion of the piston the effective force exerted is the difference between the two counteracting forces. To ascertain this, we should in this case have the diagram representing the opposing force, which was exerted

simultaneously with the force represented by diagram No. 30. That would be the diagram, the upper line of which was taken during the same stroke from the opposite end of the cylinder. We have no such diagram, but may assume it to be similar to

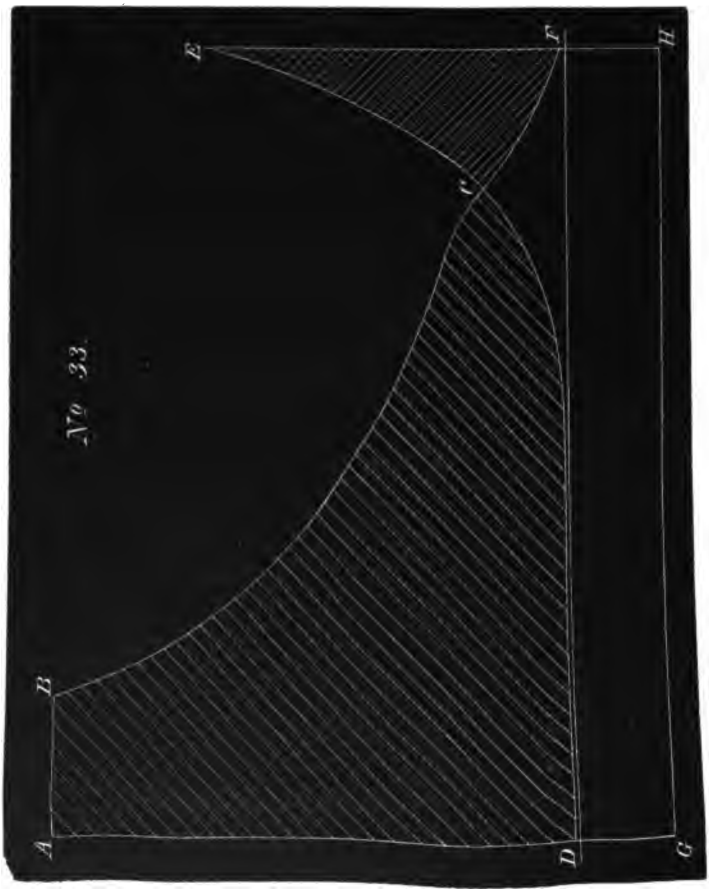
diagram No. 31. Then the figure, No. 32, will represent the force which was exerted in opposition to that shown in fig. 30.

Let us now place one of these over the other, their bases and extremities coinciding, and we obtain the figure No. 33, page 116. So far as one covers the other, the two forces neutralised each other, and may be disregarded. The projecting portion, A B C D,

of the first figure shows the force applied to the piston at each point in its stroke, up to the point C, to produce its motion, and that, C E F, of the second one, shows the force applied to it at each point beyond C to resist its motion. The two forces, G D, C F H, neutralised each other

This is the real diagram of pressure on the piston. Now it is true that, for computing the power exerted by the engine, it makes no difference from which end of the diagram the compression is deducted; and so, when one does not care to know the effective pressure on the piston to produce or to resist its motion at each point of its stroke, or the distribution of force through

the stroke, the diagram as described by the Indicator is sufficient. But in the real diagram we see in every case, also, at a glance, the total opposing forces, to what extent they neutralise each other, at what point of the stroke they are in equilibrium, and at every other point just in what degree one or the other



preponderates. This diagram ought in fact to be always drawn, for that described by the Indicator is liable to convey an erroneous impression respecting the distribution of force through the stroke, and by this means only the truth in this respect can be clearly apprehended.

A ready method of doing this is the following. Lay the diagrams from opposite ends of the cylinder one over the other, with the atmospheric lines and the extremities of the diagrams coinciding, against a window-pane. The metallic paper, to be sure, is not especially translucent, but with a bright light the under diagram can be seen, and the required line traced on the upper one with a pencil or a brass point. Every diagram should be carried down to the line of perfect vacuum; when one sees represented the real quantity of steam consumed, the quantity of heat thrown away, by being converted into forces that only counteract and neutralise each other, and the porportion which the heat so wasted bears to that which is converted into effective work.

In the discussions in Part V., whenever the diagram of pressure in the cylinder is referred to, it is the real diagram obtained in the manner above described that is always intended

PART FOURTH.

SECTION I.

OF THE CONVERSION OF HEAT INTO WORK IN THE STEAM-ENGINE.

BEFORE entering upon a critical examination of the several lines of the diagram for the purpose of getting a better understanding of the operations which are indicated by them, it seems important that we should acquaint ourselves as intimately as possible with the action which is made the subject of this section, and a general knowledge of which has indeed been already assumed.

The steam-engine is an instrument by means of which heat is converted into mechanical work. This is the operation which is ceaselessly going on, and it is the only useful action that does take place, in the cylinder of a steam-engine; while water, in its state of steam, is the medium through which this conversion is effected. The Indicator assists us in understanding this transformation, so far as we are able to understand it; and, on the other hand, without a knowledge of this action the Indicator diagram itself cannot be rightly read. To what extent this is the case, and how closely the figure described by the Indicator is connected with the action that it represents, will appear as we proceed.

At the meeting of the British Association, held at Cambridge in 1845, Mr. J. P. Joule (so reads the record—now Dr. Joule) of Manchester first exhibited the apparatus, which in its subsequently matured form has become so well known, in which he showed the temperature of water raised by the friction of a paddle-wheel, and stated the conclusion to which his experiments with it had led him: that when the temperature of 1 lb. of water is raised 1° Fahrenheit, an amount of *vis*

viva is communicated to it equal to that acquired by a weight of 890 lbs. after falling from an altitude of one foot.

At the Oxford meeting of the Association in 1847, Dr. Joule exhibited and described an apparatus, consisting of a brass paddle-wheel working in a vessel filled with liquid, with which he had repeated the experiments brought before the Association at the Cambridge meeting. By these experiments he had discovered that heat is invariably produced by the friction of fluids, in exact proportion to the force expended. Two series of experiments had been made by him, one on the friction of water, the other on the friction of sperm oil. In the former of these series the heat capable of raising the temperature of 1 lb. of water 1° Fahr. was found to be equal to the mechanical force capable of raising a weight of 781·5 lbs. to the height of one foot; whilst in the series of experiments on the friction of sperm oil the same quantity of heat was found to be equal to a mechanical force represented by 782·1 lbs. through one foot.

At the next annual meeting of the Association another paper was presented by Dr. Joule, in which he stated that since the preceding meeting a slight alteration in the form of the apparatus, calculated to give greater exactness to the results, had occurred to him, and he had commenced a new and extensive series of experiments, in order to determine the equivalent of heat with all the accuracy that its importance demands. The result arrived at, after a series of forty experiments, was an alteration of the equivalent before stated to 771, which was believed to be within $\frac{1}{200}$th of the truth. Subsequent investigations have fixed the precise equivalent of the unit of heat at 772 lbs. raised one foot. The general progress of scientific research has produced confirmation of this result, and it is now universally accepted as true, and is commonly known as Joule's equivalent. The unit of work is the work done in raising one pound through one foot, called the foot-pound, and a unit of heat, or the heat required to raise the temperature of 1 lb. of water at or near to its freezing point, 1° Fahr., is equivalent to 772 units of work.

The mechanical theory of heat is, that heat is motion, that the atoms of which all bodies consist are in continual motion, and that their absolute rest would be the absolute cold, which is considered as proven to be 461·2° below the zero of Fahrenheit, or −274 centigrade; a conclusion which was first announced by Dr. Joule, in the paper last referred to. Accord-

ing to this theory of heat, the extent and force of the movement of their atoms determines what we call the temperature of bodies. About the nature of these movements we can form only conjectures, for, like all the operations of nature, they are performed in recesses, into which we cannot hope that, while in the body, our vision shall ever penetrate. Atoms are capable of communicating their motion to adjoining atoms, of the same or of other bodies, themselves losing the amount of motion that they impart, and when increased motion is thus imparted to the atoms composing the bodies of sentient beings, the sensation is produced which we call heat, but which is in fact only an increase in the extent of the atomic motion. The enlargement of the bulk or volume of a body, which takes place when its temperature is raised, affords at once an evidence and a measure of the increase in the extent of the atomic motion.

The Generation of Steam, under constant pressure and under constant volume.

The specific heat of air under constant pressure, when it is permitted to expand freely, was determined by Regnault to be ·2377. To double the volume of 1 cubic foot of air, therefore, 9·41 units of heat need to be imparted to it.

$$\cdot 0807265 \times \cdot 2377 \times 490 \cdot 4 = 9 \cdot 41.$$

The specific heat of air under constant volume, when expansion is prevented from taking place, was ascertained by a very remote and refined investigation to be ·1678. This was only regarded, however, as a close approximation to the truth.

Then, to raise the temperature of air in the same degree as above, when prevented from expanding, there would need to be imparted to it only 6·643 units of heat.

$$\cdot 0807265 \times \cdot 1678 \times 490 \cdot 4 = 6 \cdot 643.$$

The difference is $9 \cdot 410 - 6 \cdot 643 = 2 \cdot 767$ units of heat.

When, subsequently, the mechanical theory of heat came to be established, it was observed, that when 1 cubic foot of air expanded to 2 cubic feet, 2116 foot-pounds of work were done in raising the weight of the atmosphere, so as to permit the expansion to take place.

$$14 \cdot 696 \times 144 = 2116.$$

But this required the conversion of 2·741 units of heat into mechanical work.

$$2116 \div 772 = 2 \cdot 741.$$

The co-efficient of the specific heat of air under constant volume had been ascertained so nearly, that it needed only to be increased by ·0005, or to ·1683. The mystery was solved. The simple reason of the difference in the specific heat of a gas under constant pressure and under constant volume is, that in the former case external work is done, and in the latter it is not.

The generation of steam presents a case in many respects widely different from that presented by the heating of air, though it is governed by the general law above explained. We will consider it very briefly under each of these two conditions.

It must be borne in mind that, with one seeming exception, motion is never lost, but only changes freely back and forth between atomic and dynamic modes, and must always be accounted for, in one or the other of these manifestations. This is known as the conservation of force or energy. If motion of a mass ceases, the equivalent quantity of heat must appear, and *vice versâ*.

The seeming exception is when a weight is lifted against the attraction of the earth. In this case the lifting force is in the nature of a constantly accelerating force; the same heat is converted into work that would be in imparting to the mass, if moving without resistance, the velocity that it will acquire in falling through the distance it has been lifted. The body possesses also the same mechanical energy that it would possess if so moving, and will exert it in falling to its original position. It is observed, also, that it is not warmed when its upward motion is arrested, and that in falling it does not lose heat as it acquires velocity, but that when its falling motion is arrested without being imparted to another body, heat is produced. The action is precisely what it would be if the heat that was converted into the work of lifting the body were instead converted directly into the motion it acquires in falling. This energy, suspended under the action of gravity, is called potential energy; a definition which means that we know nothing about it. We pass now to the application of the above laws to the generation of steam.

1. When steam is generated under constant pressure, the temperature of the steam already formed is not raised, nor its volume enlarged, nor its condition changed in any respect.

Successive portions of water pass into the state of steam, and equal quantities of steam before generated pass away. A portion of the heat imparted is, at the instant of the change of state, converted into the external work of overcoming the pressure, so that the enlargement of volume can take place. In the case of a steam-engine, the motion thus begun against the resistance goes on until it is imparted *to* the resistance. No change takes place here. The motion of the steam under the pressure is the same thing precisely as the motion, at the same velocity, of a mass whose weight is equal to the pressure. It is important that this should be clearly apprehended. Motion under a pressure is motion against the pressure. This is obvious, because, in order that the motion shall take place, the resistance must yield at some point, when we have pressure in motion, which is identical with weight or mass in motion. The motion imparted to the piston is communicated through the train of movements, and wherever these terminate, as well as all along the way, it appears as heat, in the increased temperature of all bodies that resist the motion.

When the steam, without having communicated motion, either atomic or dynamic, to any other body, passes directly into a condenser, the motion is arrested there, and is imparted as heat to the water or air. In this manner Regnault ascertained the heat contained in steam, permitting it to flow into a condenser, and there impart its heat to water. Of course he obtained by this method the whole of the heat, both that which, at the generation of the steam, had been converted into internal work, and that which had, at the same time, been converted into dynamic energy, or motion against resistance, and so continued up to the instant of condensation.

He maintained at all times an equilibrium of pressure between the boiler and the condenser, by pumping air into, or exhausting it from, a receiver connected with the latter; thus forming an artificial atmosphere, in which the action went on precisely as when under the atmospheric pressure. This, however, could have no influence on the result. So long as a given pressure is maintained in the boiler, the steam is formed and passes out under this pressure, and then wherever this motion ceases, and under whatever conditions, the equivalent heat must appear.

The heat converted into dynamic energy is never, strictly speaking, contained in the steam, though inseparable from its

existence when formed under constant pressure. The Tables give, of course, the total quantity of heat converted, at the generation of the steam, into both internal and external work.

2. When steam is generated under constant volume in a closed vessel, the same motion begins, at the instant of the change of state, as when it is permitted to pass away freely under constant pressure. It is formed under pressure, precisely as if that pressure *were* constant. The difference is, that the motion is arrested within the boiler, and is imparted, as heat' to the steam and water contained there—precisely as, for example, it would be imparted to the water lubricating a friction-brake. While the generation of steam in a closed boiler continues the heat is increased, not only by that imparted from the furnace directly, but also by that abandoned by the steam continually as its motion is arrested.

The quantity of heat required by steam when formed under constant volume is, then, less than when formed under constant pressure, by the quantity equivalent to the external work done in the latter case on change of volume, precisely as in the case of a permanent gas.

Dynamical energy is exerted by steam, or motion is imparted by it to matter, when the changes of state take place from water to steam, and again from steam back to water. Steam exerts no force, except as the accompaniment of one or the other of these changes of state. The first change involves an enlargement of volume, and the overcoming of whatever resistances may oppose such enlargement; the second change involves a loss of atomic motion, which motion the steam can lose only by imparting it to other bodies; and it may then appear first in the form of increased atomic motion, or heat, of bodies in contact, and then in the motion of their mass through space. This action we will now endeavour to make clear.

When 1 lb. of water at the temperature of 212°, and containing 212·9 thermal units, passes, under the pressure of the atmosphere, into the state of steam of the same temperature, 965·7 units of heat disappear. These are equivalent to 745,520 units of work; and they are, in fact, wholly converted into internal and external work. The latter is the work done in overcoming the resistance which the atmosphere opposes to the expansion of the water into steam, and the amount of it we are able to fix exactly.

The atmosphere exerts a pressure of 2116·22 lbs. on a square foot. One pound of water, when converted into steam under this pressure, expands to occupy a space of 26·336 cubic feet. The space at first occupied by the water was ·016 of a cubic foot, so that the space from which the atmosphere is expelled is 26·32 cubic feet. To do this the steam must perform 2116·22 × 26·32 = 55,700 foot-pounds of work. This is equivalent to 72·15 units of heat, which being deducted from the 965·7 units which have disappeared, leaves the great amount of 893·55 units, which have been converted into the increased internal work, amounting to 689,820 foot-pounds, which the atoms of the vapour ceaselessly perform, in vibrating or otherwise moving through spaces having an amplitude 1644 times greater than those through which they were moving when in the state of water.

It is important that the distinction between internal and external work should be clearly apprehended. Without expansion, it is the latter only that can be utilised, being all the dynamical energy that is exerted on the change of state from water into steam. Let us endeavour to follow the process with some care, and observe its nature and limitations. For this purpose we will suppose a vessel of sufficient height, and having a cross area equal to one square foot, in which we place 1 lb. of water, and above it a piston, which is conceived to be without weight, and to move in the vessel steam-tight without friction. We will suppose, also, that the heat applied is received and retained by the water and the steam without loss. These are, of course, purely theoretical conditions. We shall see afterwards how far the results are modified by the conditions which are encountered in practice.

First.—The piston rests upon the water, and upon the piston rests the atmosphere, constituting a weight of 2116·22 lbs. Let the water have already a temperature of 212°, which under this pressure cannot be exceeded. As now additional heat is imparted to the water, corresponding portions of the latter are successively converted into steam, and the piston is raised. When 965·7 units of heat have been so imparted, the piston stands at an elevation of 26·336 feet above the bottom of the vessel, and the last drop of water has disappeared, the space below the piston being occupied by a perfectly transparent and invisible gas. The external work done has been moving the piston through a distance of 26·32 feet against a resistance of

2116·22 lbs., amounting to 55,700 foot-pounds of work, representing 72·15 units of heat, or ·0747 of the heat imparted. This is the full theoretical efficiency of saturated steam at the atmospheric pressure, when not worked expansively. When the temperature of the water has first to be raised to 212° from some lower point, the efficiency is by so much diminished.

There is no way except by expansion, of which we will speak presently, in which the above proportion of the heat theoretically utilised can be increased. To make this clear:

Second.—Let our piston be loaded with a weight, including the atmosphere, of 90 lbs. on the square inch, or 12,960 lbs. on the square foot, and let the water have a temperature of 320·039°, which is its boiling-point under this pressure. We will now impart to it 888·34 additional units of heat. While these are being received the piston is rising and the volume of water is diminishing. When all have been imparted the water will have disappeared, the piston will stand at an elevation of 4·73 feet above the bottom of the vessel, and the space under it will, as before, be occupied by invisible vapour; which has, however, a volume only 294·7 as great as the water from which it has been evaporated. The external work done has been 12,960 × 4·73 = 61,300 foot-pounds, or 5600 foot-pounds more than in the former case, while the heat imparted has been 77·36 units less. The proportion of the heat imparted which has been utilised is ·089, and this is the full theoretical efficiency of saturated steam at 90 lbs. pressure when not worked expansively. A comparison of the two cases is as follows:—

	Units.
In the first place:—Heat disappeared	965·7
Of which there were converted into external work	72·15
„ „ converted into internal work	893·55
In the second place:—Heat disappeared..	888·34
Of which there were converted into external work	78·98
„ „ converted into internal work	809·36

The apparent gain in the second case is as follows:—

No. of units of heat required to be imparted is less by	77·36
„ „ utilised is greater by	6·83
Total apparent gain	84·19

But the additional heat required in the second case to be imparted to the water, to raise its temperature to the boiling-point, is 323·176 − 212·9 = 110·276 units. There is, therefore, in this case a net loss of 110·276 − 84·19 = 26·086 units. Practically, however, and without reference to its value for the purpose of expansion, there is a great gain in employing steam of a high pressure. Small as is the proportion of the heat that is theoretically utilised when steam is at the atmospheric pressure, even this is entirely unavailable unless the resistance of the atmosphere be first removed. Then, to the extent that this has been done, other resistances may be substituted for it. But pressures above that of the atmosphere are, under all circumstances, wholly available for useful work, and the higher the pressure the greater the amount of force that can be utilised, relatively to that which must be thrown away.

Even in condensing engines, considering that the water must enter the boiler at temperatures not much exceeding 100°, that three or four pounds of pressure on the square inch must be thrown away, and that much heat must be expended in maintaining the temperature of the cylinder, &c., it is evident from the above analysis that, without expansion, not more than five per cent. of the heat imparted to the water becomes converted into mechanical work.

Before passing from this branch of our subject, let us refer again for a moment to the nature of the action, as above illustrated. The work we have been considering is performed entirely in the boiler, in the expansion that takes place there on the change of state from water to steam, and is thence transmitted to the piston of the engine. A given density of steam being maintained, all evaporation in the boiler, after heat lost in any manner has been restored, must be represented by a corresponding motion of the piston; and although the restoration of lost heat involves condensation of the steam, the motion of the piston thus produced does not involve any such condensation.

Instead of our piston in the preceding illustration being raised, against the resistance of the atmosphere, to a height of 26·32 feet, let us suppose it to be fixed at that height, and conceive the space below it to be an absolute vacuum. In this space the pound of water is evaporated, exactly filling it with steam, of the same pressure, or elastic force, with the atmosphere. Let the piston be now released. It remains motion-

less, the forces above and beneath it being in equilibrium. Everything is in the same condition as before supposed.

But the piston had been lifted, against the weight of the atmosphere, to its present position by other forces before the steam was generated. The evaporation has gone on under constant volume, and the heat required to effect it has been less than in the first case supposed, by the quantity then required to be converted into this external work. This quantity we have seen to be 72·15 units, leaving only 893·55 units necessary now to be imparted. This, therefore, is all the heat that the steam now contains. But it is in the same precise state that it was in the former case, filling the same space, under the same pressure.

We conclude, then, that the heat contained in steam that has done external work—all the heat which the steam can properly be said to contain—is less than the total heat given in Tables III., IV., and V., by the quantity which is the equivalent of such external work.

It has been deemed important to present a Table of this heat —the heat contained in steam which has not been expanded, and that is thrown away except as a portion of it may be converted into work by expansion.

The heat converted into work on the evaporation, and represented in a steam-engine in piston-motion up to the point of cut-off, is found by the following formula:—

Let v = the relative volume of the steam;
p = its elastic force, in lbs. on the square inch; and
T = the thermal units converted into external work in the boiler;

Then $T = \cdot 002988\, p\, (v - 1)$.

The relative volumes not being known with exactness, it would be idle to go in this Table beyond the second place of decimals.

There is one way only in which the enormous loss of heat shown above can be diminished. As water in passing into the state of steam affords, by its expansion to the volume it must occupy, the power we have been considering, so steam, in returning to the state of water, may be made to impart a portion of its atomic motion to the piston. If the piston were colder than the steam, the latter would condense, not to move, but to warm the former; the atomic motion would be lost by the steam, only to appear as atomic motion of the metal. This

Table VIII.

Showing the heat converted into external work, on the evaporation of 1 lb. of steam, under each lb. of pressure, from 1 lb. to 210 lbs. on the square inch.

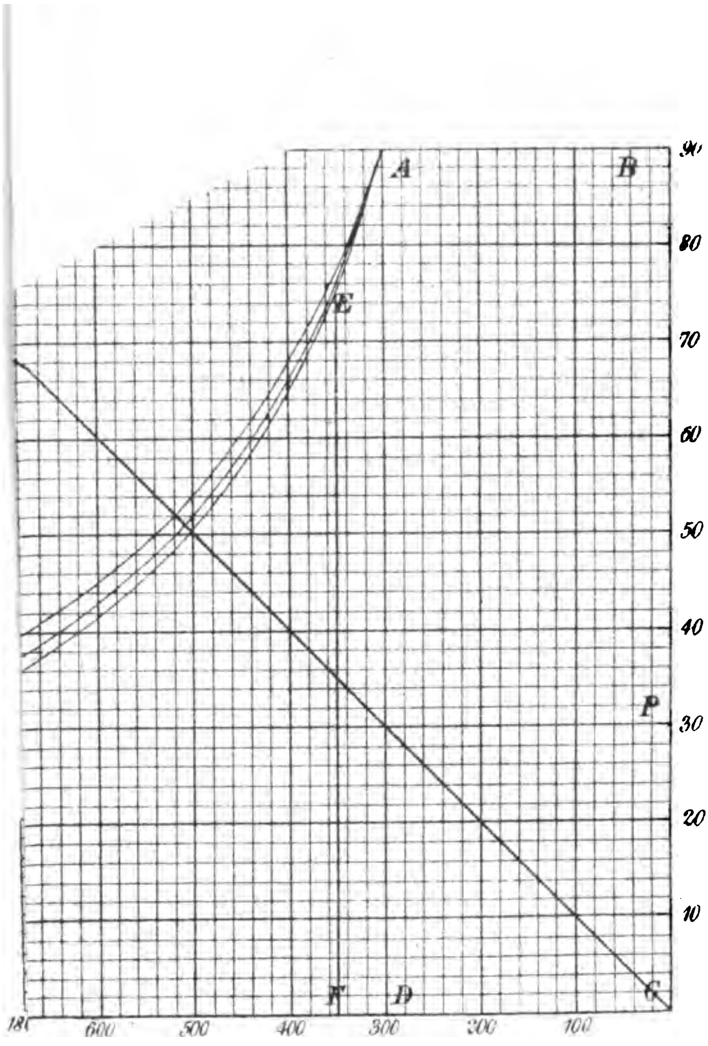
Pressure, from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.	Pressure, from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.
1	61·62	1083·41	34	74·42	1118·05
2	64·07	1088·38	35	74·50	1118·49
3	65·67	1091·46	36	74·60	1118·89
4	66·92	1093·70	37	74·70	1119·29
5	67·76	1095·69	38	74·78	1119·69
6	68·37	1097·46	39	74·86	1120·09
7	69·00	1098·90	40	74·93	1120·48
8	69·54	1100·19	41	75·00	1120·85
9	70·09	1101·28	42	75·09	1121·22
10	70·49	1102·39	43	75·20	1121·55
11	70·87	1103·39	44	75·30	1121·88
12	71·25	1104·29	45	75·41	1122·19
13	71·68	1105·05			
14	72·00	1105·86	46	75·50	1122·51
15	72·21	1106·70	47	75·58	1122·84
			48	75·66	1123·16
16	72·34	1107·57	49	75·75	1123·46
17	72·43	1108·43	50	75·82	1123·78
18	72·56	1109·20	51	75·90	1124·08
19	72·73	1109·90	52	75·98	1124·37
20	72·86	1110·59	53	76·06	1124·66
21	72·98	1111·27	54	76·13	1124·95
22	73·10	1111·91	55	76·21	1125·23
23	73·27	1112·47	56	76·30	1125·50
24	73·36	1113·09	57	76·38	1125·76
25	73·49	1113·65	58	76·47	1126·02
26	73·58	1114·22	59	76·56	1126·26
27	73·66	1114·79	60	76·64	1126·52
28	73·79	1115·28			
29	73·92	1115·75	61	76·73	1126·76
30	74·04	1116·22	62	76·82	1127·00
			63	76·90	1127·23
31	74·10	1116·74	64	76·97	1127·48
32	74·20	1117·19	65	77·05	1127·71
33	74·32	1117·62	66	77·13	1127·94

TABLE VIII.—*continued.*

Pressure, from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.	Pressure from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.
67	77·20	1128·17	106	80·24	1134·90
68	77·28	1128·40	107	80·31	1135·04
69	77·36	1128·61	108	80·39	1135·17
70	77·43	1128·84	109	80·47	1135·29
71	77·52	1129·04	110	80·53	1135·44
72	77·60	1129·25	111	80·60	1135·57
73	77·68	1129·45	112	80·67	1135·71
74	77·75	1129·66	113	80·74	1135·84
75	77·83	1129·88	114	80·80	1135·98
			115	80·86	1136·11
76	77·92	1130·05	116	80·94	1136·23
77	78·00	1130·23	117	81·02	1136·35
78	78·09	1130·42	118	81·10	1136·46
79	78·17	1130·61	119	81·18	1136·57
80	78·24	1130·80	120	81·25	1136·69
81	78·32	1130·98			
82	78·40	1131·16	121	81·34	1136·79
83	78·47	1131·35	122	81·43	1136·89
84	78·53	1131·54	123	81·52	1136·99
85	78·60	1131·73	124	81·60	1137·10
86	78·68	1131·89	125	81·69	1137·19
87	78·75	1132·07	126	81·77	1137·30
88	78·83	1132 24	127	81·84	1137·41
89	78·91	1132·40	128	81·92	1137·51
90	78·98	1132·57	129	82·00	1137·61
			130	82·07	1137·72
91	79·06	1132·73	131	82·15	1137·82
92	79·14	1132·89	132	82·23	1137·92
93	79·22	1133·04	133	82·31	1138·02
94	79·29	1133·20	134	82·38	1138·12
95	79·37	1133·35	135	82·45	1138·23
96	79·45	1133·50			
97	79·52	1133·66	136	82·53	1138·32
98	79·60	1133·80	137	82·60	1138·42
99	79·68	1133·95	138	82·68	1138·51
100	79·75	1134·10	139	82·76	1138·60
101	79·84	1134·23	140	82·83	1138·70
102	79·93	1134·36	141	82·91	1138·79
103	80·00	1134·49	142	83·00	1138·87
104	80·08	1134·64	143	83·07	1138·97
105	80·16	1134·77	144	83·12	1139·08

TABLE VIII.—*continued.*

Pressure from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.	Pressure from the absolute vacuum, in lbs. on the square inch.	Thermal Units converted into external work on the evaporation.	Thermal Units actually contained in the steam.
145	83·19	1139·18	178	85·62	1141·75
146	83·28	1139·25	179	85·68	1141·83
147	83·36	1139·34	180	85·72	1141·93
148	83·44	1139·42	181	85·82	1141·97
149	83·52	1139·50	182	85·92	1142·01
150	83·59	1139·59	183	86·00	1142·07
151	83·67	1139·67	184	86·09	1142·11
152	83·75	1139·75	185	86·18	1142·16
153	83·83	1139·83	186	86·25	1142·22
154	83·89	1139·92	187	86·31	1142·30
155	83·96	1140·01	188	86·38	1142·37
156	84·03	1140·10	189	86·45	1142·43
157	84·09	1140·19	190	86·51	1142·50
158	84·16	1140·27	191	86·59	1142·56
159	84·23	1140·36	192	86·67	1142·61
160	84·28	1140·46	193	86·75	1142·66
161	84·37	1140·52	194	86·82	1142·72
162	84·46	1140·58	195	86·88	1142·79
163	84·55	1140·64	196	86·97	1142·83
164	84·63	1140·71	197	87·05	1142·88
165	84·71	1140·78	198	87·14	1142·92
166	84·78	1140·86	199	87·23	1142·96
167	84·84	1140·95	200	87·31	1143·01
168	84·91	1141·02	201	87·39	1143·06
169	84·98	1141·10	202	87·46	1143·11
170	85·03	1141·19	203	87·53	1143·17
171	85·12	1141·25	204	87·59	1143·24
172	85·21	1141·30	205	87·66	1143·29
173	85·29	1141·37	206	87·74	1143·33
174	85·37	1141·43	207	87·81	1143·39
175	85·45	1141·49	208	87·89	1143·44
176	85·51	1141·58	209	87·97	1143·48
177	85·56	1141·67	210	88·03	1143·55



A
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180
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300
200
100

demand must first be satisfied before any dynamical results can be obtained. No piston can be put in motion by confined steam, unless the temperature of its surface, as well as of all the surfaces in contact, is the same as that of the steam. Then the motion lost by the steam, and which cannot go to augment further the atomic motion of the metal, appears as motion of its mass. This temperature of the surfaces being maintained or restored, the atomic motion of the steam becomes converted into motion of the mass of the piston whenever the pressure of the steam falls, and it expands to a larger volume.

This expansion may take place under various conditions. One extreme condition is, when all the steam contained in the cylinder and boiler and connections expands, or suffers fall of pressure, together; the opposite extreme condition would be, if at the commencement of each stroke a portion of the steam, sufficient only to fill the piston displacement for a very small part of its stroke, without any waste room, were confined in the cylinder, by the communication with the boiler being then cut off, and were permitted to expand during the remainder of the stroke to an extreme tenuity at its termination. Under the first of these conditions the expansion during a single stroke would be inappreciable, and so of no value; under the second it would cause the utmost proportion of the atomic motion of the steam to be converted into piston motion. The economical problem is, how in practice to realise most nearly the gain which in theory is thus effected.

For the examination of the action of the steam in expanding we will now return to our vessel, which, under the conditions supposed, we shall find in precisely the state in which we left it. The source of heat was removed from it when the evaporation was completed, and we will suppose that no further heat is imparted to the steam, nor any lost by it except through its conversion into work. Diagram No. 34 (in lithograph) has been drawn to exhibit this expansive action, and to it, as his guide, the reader will now be introduced.

On this diagram the distance B A represents the height, 4·73 feet, to which our piston was raised by the evaporation, and at which it has since been standing, and also the volume, 4·73 cubic feet, of 1 lb. of steam, under the pressure of 90 lbs. on the square inch; and the rectangle A B C D represents the work done by a pressure of 90 lbs. on the square inch acting through the above distance, as already given.

From the point A three curves are drawn. The upper one, A G, is a hyperbola. This curve represents expansion according to the law of Mariotte, which is, as has been already explained, page 52, that the density and the volume of a gas vary inversely as each other. C N is the axis of this curve, and C B and C H are its asymptotes. The former of these is the line which the compression of the steam would reach, and the latter that which its expansion would reach, if these were continued to an infinite distance. The corresponding portions of the curve on opposite sides of the axis C N are, of course, the counterparts of each other, as may be better observed on diagram No. 38, in the next section, p. 161.

The product of an ordinate of the asymptote C H, and its corresponding abscissa, representing, the former the density and the latter the volume of the steam, is a constant number, to whatever point of the curve these may be drawn. This is an isothermal curve, the law of Mariotte assuming a constant temperature of the gas, and so it cannot be the curve of the expansion of steam; but it affords the base of measurement and the standard of comparison for all other expansion curves theoretical and actual.

The second curve, A I, is the curve of the relative volumes of steam, according to the generally received speculations, and which volumes have been given in the Tables of the Properties of Steam, pages 58 and 66. We see at a glance in what degree steam departs, or is believed to depart, from the law of the gases; expanding considerably less than that law calls for on a given fall of pressure, but not so much less as it would do if it conformed to the law of the contraction of air by cooling. The contraction from the Mariotte curve which this law would require would, for the fall of temperature here represented, be 201·6 from 1800; that given in the Table of Relative Volumes, and shown by the curve A I, is 159 from 1800.

Theoretically, the curve A I could not be drawn, because in expansion work must be done, and so steam be condensed. This curve consists, then, merely of the separate points to which a given weight of water would expand on evaporation, if evaporated under different pressures represented by the ordinates.

The lower curve, A L, represents the theoretical expansion of steam; its volume being reduced continually by the condensation consequent on the conversion of successive portions of its

TABLE IX.

EXPLANATORY, IN PART, OF DIAGRAM No. 34.

Showing the work done by one pound of steam, of 90 lbs. pressure, in raising a piston having an area of one square foot, supposing it to expand on the curve A G to the different points indicated, according to the Law of Mariotte.

		1.			2.			3.		
Proportions of the rise at which the steam is cut off.	Terminal pressures, in lbs. on 1 square inch.	(a) Mean pressures during the rise, in lbs. on 1 square inch.	(b) Total distance through which the piston is raised, in feet.	(c) Total work done in foot-pounds.	(a) Mean pressures during the expansion in lbs. on 1 square inch.	(b) Distance through which the piston is raised by the expansion, in feet.	(c) Work done during the expansion in foot-pounds.	(a) Mean pressure during each fall of 5 lbs. by expansion, in lbs. on 1 square inch.	(b) Distance which the piston is raised during each of these intervals in feet.	Work done during each of these intervals in foot-pounds.
1·0000	90	90·	4·730	61,300						
·9444	85	89·85	5·008	64,796	87·35	·278	3,500	87·35	·2783	3,500
·8888	80	89·40	5·321	68,500	84·60	·591	7,200	82·35	·3131	3,710
·8333	75	88·65	5·676	72,457	81·90	·946	11,157	77·35	·3548	3,950
·7777	70	87·605	6·081	76,712	79·22	1·351	15,412	72·35	·4052	4,220
·7222	65	86·12	6·550	81,228	76·03	1·820	19,928	67·35	·4683	4,540
·6666	60	84·30	7·095	86,126	72·90	2·365	24,826	62·35	·5456	4,898
·6111	55	82·048	7·740	91,448	69·55	3·010	30,148	57·35	·6449	5,325
·5555	50	79·40	8·514	97,345	66·15	3·784	36,045	52·35	·7742	5,835
·5000	45	76·185	9·460	103,781	62·37	4·730	42,481	47·35	·9460	6,450
·4444	40	72·44	10·642	111,010	58·40	5·912	49,710	42·35	1·1825	7,210
·3888	35	68·04	12·162	119,160	54·06	7·432	57,860	37·35	1·5199	8,174
·3333	30	62·97	14·190	128,670	49·455	9·460	67,370	32·35	2·0276	9,445
·2777	25	57·025	17·028	139,827	44·34	12·298	78,527	27·35	2·8379	11,175
·2222	20	50·08	21·285	153,497	38·67	16·555	92,197	22·35	4·2569	13,700
·1666	15	41·88	28·380	171,152	32·256	23·650	109,852	17·35	7·0948	17,725
								Totals—	23·6500	109,857

TABLE X.—COMPLETING THE EXPLANATION OF DIAGRAM NO. 34.

In this Table the lengths of the abscisses of the three curves, measured from the line B C, *and representing the volume of the steam, are given, at the end of each interval of the expansion taken, according to the scale of the diagram. The powers exerted by the steam during each of these intervals, in expanding on the curves* A I *and* A L, *are derived from those exerted in expanding on the curve* A G, *as given in the preceding Table, by making the reductions corresponding with the diminutions in the lengths of the parts of the abscisses which represent the expansion.*

Terminal pressures, lbs. on 1 square inch.	1. Lengths of the abscisses of the Mariotte curve, A G.	2. Lengths of the abscisses of the curve of relative volumes, A I.		3. Lengths of the abscisses of the expansion curve, A L.			4. Foot-pounds of work done during each interval, expanding on the curve A I.	5. Foot-pounds of work done during each interval, expanding on the curve A L.
			Diminution from curve A G.		Total reduction of volume by condensation.	Reduction of volume in each interval by condensation.		
90	300·	300·	(a)	300·	(a)	(b)		
85	317·65	316·	1·65	315·06	·94	94	3,173	2,986
80	337·5	334·2	3·3	332·20	2·00	1·06	3,384	3,186
75	360·	354·5	5·5	351·32	3·18	1·18	3,588	3,378
70	385·7	377·9	7·8	373·35	4·55	1·37	3,836	3,612
65	415·385	404·8	10·585	398·67	6·13	1·58	4,123	3,882
60	450·	436·2	13·8	428·23	7·97	1·84	4,447	4,187
55	490·9	473·	17·9	462·87	10·13	2·16	4,826	4,541
50	540·	517·6	22·4	504·85	12·75	2·62	5,290	4,980
45	600·	571·9	28·1	555·93	15·97	3·22	5,846	5,502
40	675·	639·5	35·5	619·50	20·00	4·03	6,527	6,143
35	771·4	726·2	45·2	701·00	25·20	5·20	7,391	6,956
30	900·	841·7	58·3	809·47	32·23	7·03	8,527	8,020
25	1080·	1002·5	77·5	960·29	42·21	9·98	10,065	9,460
20	1350·	1242·	108·	1184·67	57·33	15·12	12,291	11,543
15	1800·	1641·	159·	1557·63	83·37	26·04	15,846	14,194
						Totals ..	99,160	92,570

latent heat into mechanical work. The shortening of its abscisses from those of the curve A I represents this contraction of volume. The method by which this has been determined will be described directly.

The Table of Hyperbolic Logarithms enables us to calculate the power exerted during the expansion on the curve A G only. In Table No. IX., this action will be found fully presented. In the succeeding Table, No. X., the relative lengths of the abscisses of the three curves are given for each of the sixteen points taken, and also the powers exerted by the steam in expanding on the curves A I and A L, as reduced from those exerted by expansion on the curve A G, according to the diminution in the lengths of those parts of the abscisses which represent the expansion.

Let now one-sixth of the weight resting on our piston, or 2160 lbs., be removed, in a manner similar to that in which pressure falls by expansion, as shown by the portion A E of the curve A I. The pressure is reduced from 90 lbs. to 75 lbs. on the square inch. If no condensation took place, the volume of the steam would enlarge from 294·7 times to 348·3 times that of the water. But to effect this enlargement, a mean weight of 11,790 lbs. must be lifted through ·86 of a foot, which is 10,145 foot-pounds of work, and this would require the conversion of 13·14 units of heat into mechanical work. Of this number the diminution in the total heat of the steam supplies 3·86 units, leaving 9·28 units which must be supplied by the condensation of a portion of the steam.

The steam condensing retains still the same temperature, and so parts only with its latent heat. The whole number of units of latent heat contained in 1 lb. of steam, at 75 lbs. pressure, is 819·796; so that nearly $\frac{1}{90}$ of the steam must be condensed to supply this power, and as the piston rises we see the interior of our vessel become cloudy from this condensation. After the work has been performed it will gradually clear again, the water formed collecting at the bottom.

This condensation, however, reduces by so much the volume of the steam, and therefore the rise of the piston, and less work is done, and again the condensation is not so much. The actual rise of the piston is found, by the method to be explained presently, to be ·81 of a foot; the work done, 9550 foot-pounds; and the heat converted into work, 12·37 units: all which is represented on the diagram in the area A E F D.

Again, instead of removing one-sixth, let us, in the same gradual manner in which the pressure would fall by expansion, without allowing for condensation, as represented by the curve A I, remove five-sixths of the load on our piston, amounting to 10,800 lbs., leaving 2160 lbs. remaining at the end of the expansion.

The pressure now falls to 15 lbs. on the square inch, and if no condensation took place the volume of the steam would expand from 294·7 times to 1612 times that of the water, causing a rise of the piston, additional to the 4·73 feet produced by the evaporation, of 21·15 feet, and raising a mean weight of 4688 lbs. through this distance, and so exerting 99,160 foot-pounds of power.

We are struck with astonishment at the amount of this power. The power developed by the evaporation was 61,300 foot-pounds. The additional power, derived from cutting off at five-sixths, and expanding though one-sixth, of the rise, was found to be, apparently, before the deduction had been made for condensation, 10,145 foot-pounds. Now we are, in round numbers, cutting off at one-sixth, and expanding through five-sixths, of the rise; and although at the end the pressure has fallen five times as far as it then did, still the work done on this expansion would, if no deduction had to be made for condensation, be 99,160 foot-pounds, or very nearly ten times as much as on the former one. The explanation of this is found in the comparatively great distance through which the piston would move on this expansion, 21·15 feet, against only ·86 of a foot on the former one. If the expansion followed the law of Mariotte, 25 times the distance would be moved through to effect a five-fold expansion: the above distances are 24·6 and 1. It is interesting to observe in the Tables and on the diagram precisely how, as the mean pressure falls, the amount of work done on a given fall of pressure increases.

But to lift this weight through this distance, 128·45 units of heat would have to be converted into mechanical work. The total heat of the steam falls from 1211·552 units to 1178·912 units, so that 32·64 units are set free, or, as happily expressed by M. Regnault, are abandoned, by the steam, and supply this requirement to that extent. There would remain then 95·81 units, which must be furnished by the condensation of a portion of the steam. The quantity of latent heat contained in the steam at 15 lbs. pressure is 892·76 units, so that ·1073 of the

pound of steam must be condensed; and now, as our piston rises, we see in the vessel not only a thick cloud, but a shower. When this has cleared away we find that the piston has risen, not 21·15 feet, but only 19·835 feet.

The method of arriving at this distance is as follows:—A single computation would give a widely eroneous result, from the fact that the rate of condensation is not uniform during the expansion. The computation has been made, therefore, for each 5 lbs. fall of pressure, and the result approximates demonstrably very nearly to the truth.

The process for the first interval is given as an illustration. The power exerted by the steam in expanding from 90 lbs. to 85 lbs. on the curve A I would be 3173 foot-pounds, requiring the conversion of 4·11 units of heat into work. 1·2257 units of heat are abandoned by the steam, so that 2·8843 units must be supplied by condensation. This produces a contraction of ·00324 of the volume, or of 1·0238 on the scale of the diagram and of Table X., and reduces the distance raised by expansion from 16· to 14·9762. But the same weight lifted through the latter distance would require only 2970 foot-pounds of power to be exerted, and the condensation produced in exerting this power would reduce the distance raised by expansion only to 15·071. We have now three expansions and their differences:—

$$\begin{array}{ll} 16\cdot0000 & \\ & -\ 1\cdot0238 \\ 14\cdot9762 & \\ & +\ \ \cdot0948 \\ 15\cdot0710 & \end{array}$$

It will not be necessary to carry the calculation farther, as the differences in alternate directions will continually diminish by the same divisor, which in this case is 10·8; and we may continue the above series, as follows:—

$$\begin{array}{ll} 15\cdot0710 & \\ & -\ \cdot0088 \\ 15\cdot0622 & \\ & +\ \cdot0008 \\ 15\cdot0630 & \end{array}$$

The actual contraction of volume by condensation is, therefore, 16· − 15·063 = ·937.

This calculation is repeated for each interval; the proportion of loss is found each time, be dividing the units of heat con-

densed by the units of latent heat remaining, and the expanded volume, as reduced by previous condensation, is then multiplied by this decimal. In this manner column 3 (*b*) of Table X. has been found, and from this the other columns relating to the curve A L have been derived.

On this expansion 92,570 foot-pounds of work have been done, and 120 units of heat have been converted into work, of which number 87·36 units have been supplied by the condensation of ·0978 of the steam.

The total result is as follows:—

	Foot-pounds of work done.	Units of heat converted.
In the change of state:—		
1st. From water into steam ..	61,300	79·4
2nd. From steam into water ..	92,570	120·
Total	153,870	199·4

The excess of power furnished by the condensation over that furnished by the evaporation is 31,270 foot-pounds. The total number of units of heat above 212·9 imparted to 1 lb. of water to evaporate it under 90 lbs. pressure is 998·625; and of this number we see that, when expanding through five-sixths of the rise, 199·4, or, as nearly as possible, ·2, is converted into mechanical work.

A point may be noted here with respect to expansion on the Mariotte curve, which, as we shall have occasion to see, is the lowest curve that is reached in practice. If steam is used at any lower pressure than 90 lbs., for example, and follows the piston for a proportionately longer distance before it is cut off —as, if one-half the pressure follows twice as far, or if one-third of the pressure follows three times as far—the expansion will describe the remaining part of the same curve.

From this it follows, that the additional power obtained by carrying a higher pressure and cutting off correspondingly sooner is clear gain; as, for example, the power represented by the area A O P B, diagram No. 34, is all gained by cutting off 90 lbs. at one-sixth, instead of 30 lbs. at one-half, of the stroke.

So again, if steam were used of 180 lbs. pressure, and were cut off at one-twelfth of the stroke, an area corresponding with O G H R would be added, above the line A B, to the power exerted by the same weight of steam.

We have observed that, theoretically, the curve A I is impossible. We have now to remark that the curve A L is practically impossible; because, as the pressure and temperature fall by the expansion, the steam condensed each instant is re-evaporated the next. The heat it has parted with, however, is gone—converted into piston motion. There is not required so much to re-evaporate it, its total heat at the end of the expansion being, as already seen, 32·64 units less than at the commencement.

As the boiling-point falls very rapidly, falling from 320·039° to 213·025°, during this expansion, the heat required to re-evaporate the water as it is formed is supplied partly from its own excess of heat, but mostly from the surfaces of the cylinder piston, and heads, from which the heat is abstracted by the steam with enormous rapidity. Thus we see that the curve A I, although theoretically impossible, is practically the lowest curve that can be drawn, and that the heat converted into work during the expansion, above that abandoned by the steam, is represented at the end of the stroke by a lower temperature of the confining metal, instead of by water formed. We note also, that the work done during expansion is not subject to deduction for contraction of volume of the steam by condensation, and cannot be less than that given in column 4 of Table X.

We come now to the subject of condensation in the cylinder. Let us suppose diagram No. 34 to represent the action of steam, cut off at one-sixth of the stroke, in the cylinder of an engine, and that at the point A the cylinder is filled with perfectly dry steam. During the expansion the temperature of the interior surfaces falls nearly with that of the steam, affording heat to re-evaporate the water formed, and also to superheat the steam, perhaps considerably, as the metal contains heat in excess. During the exhaust the steam, supposing it not to fall to any lower temperature than that which it reached at the end of the expansion, nevertheless continues to abstract heat from the metal, as this is brought from the body to the surface by conduction. These conditions involve the smallest loss of heat possible.

The heat thus lost *must* be restored to the metal before the piston arrives again at the point A. We have already observed that, until the exposed surfaces of the metal have acquired a temperature equal to that of the steam, the heat in the steam goes to warm the piston, and not to move it. There are three ways of restoring the heat which the surfaces have lost,—first, by superheating the steam, so that it can impart the lost heat to the cylinder without suffering any condensation, and this is probably the only mode by which the condition we have supposed, of entirely dry steam at the point of cut-off, can be realised; second, by using a steam-jacket; and third, by permitting the entering steam to suffer the condensation necessary for this purpose.

The latter course is the one commonly adopted. Let us follow the action in such a case, as it takes place on one side of the piston, supposing the steam to enter the cylinder perfectly dry, and supposing also that, on the first stroke we are observing, the cylinder is filled with perfectly dry steam at the point of cut-off. The condensation, and cooling of the cylinder by re-evaporation, on the first expansion, we have just observed. At the commencement of the next stroke a part of the entering steam must be condensed to restore this heat, so that at the point of cut-off water is present, representing the work done on the previous expansion and the heat abstracted by the steam during the exhaust.

As from this point the temperature of the steam, and the boiling point, again fall rapidly, the water takes the excess of heat from the surfaces with the greatest avidity, so that at the end of the second expansion it, as well as the water formed by the work done, is all evaporated; but the surfaces are cooled twice as much as they were at the end of the previous stroke, even disregarding the heat which they lost by being exposed to the cooler steam during the first exhaust. Thus at the commencement of the next stroke twice as much of the entering steam must be condensed on them to restore the heat they have lost, and then this double quantity of water formed must be evaporated during the expansion and return stroke, and so on.

But where does this process stop, and why does it not go on till all the entering steam has to be condensed? It does go on much farther than is commonly imagined. First, the steam ceases to be superheated at the end of the expansion, and after

a while the moisture formed fails to be wholly re-evaporated, though this process continues through the return stroke, and then, and not until then, the increase in the condensation can cease.

Clearly, so long as the surfaces afford heat to re-evaporate all the water formed, an additional quantity of water must be formed at the beginning of each successive stroke; but when they cease to do this, a point is reached below which they are not cooled, and so a limit is formed to the condensation required to restore their temperature. This limit will, it is clear, vary as the extent of the exposed surfaces, and as the duration of the exposure; and the proportion of the steam that is condensed will be inversely as the weight admitted to the cylinder. So the loss from this cause must be greatest: first, in the smallest cylinders with the shortest stroke, these having the largest proportion of surface to capacity; second, in engines having the slowest moving pistons; and third, when the steam is cut off earliest. Concerning high-speed engines it is to be observed, that they are more economical than slow-moving engines working steam of the same pressure, and cutting off at the same point; not in the degree that their piston speed is greater, but as their area of surface, exposed to the same weight of steam in a given time, may be less; which is not necessarily, nor always, the case. If it were attempted to make a cylinder of metal having greater heat-conducting power than that of iron—as brass, for example—the loss from this cause would be found to be increased directly as the rate of conduction was increased.

We have supposed the steam to enter the cylinder dry. In fact, however, unless superheated, it never is dry, and often contains a great amount of water. But if the condensation of dry entering steam goes on increasing, so long as the cylinder can, during the expansion and return stroke, re-evaporate the water formed, as it certainly must do, it does not appear how water contained in the steam and diffused as a cloud through it can increase this action materially; and so it would seem that the loss occasioned thereby is mostly that of the heat carried away from the boiler by the useless water.

The condensation of the entering steam can be readily observed. An indicator stop-cock on the steam-chest and those on the cylinder being opened, a marked difference will be seen in the appearance of the escaping steam, that from the cylinder

being much the more cloudy. Dry steam is invisible; very wet steam, on the other hand, sometimes blows through the stop-cock in appearance like a stem of white glass. The water formed by condensation must of course be at first deposited on the cooler surfaces of the metal, but the violence of the impact and current seems instantly to detach the particles, and diffuse them through the body of the steam.

A simple experiment will beautifully exhibit the evaporation which takes place during expansion. When an engine is stopped, without allowing the cylinder to cool, block the fly-wheel and turn on the steam. This being then shut off again, the cylinder at one end is filled with confined steam of the boiler pressure. Now partially open the Indicator stop-cock, and allow the steam to blow out gradually. At first it will blow more or less cloudy; as the pressure falls by expansion, and the force of the current slackens, the colour will be gradually disappearing, and before ceasing to escape the steam will have become quite invisible. Here time is allowed for the evaporation to be completed. A like observation may be made while the engine is running, when the evaporation will be seen to be only partial, the time occupied in a stroke being too short for its completion, unless the steam enters the cylinder dry. Very different results will be obtained in different cases, according to the proportion of water in the steam, the pressure carried, the point of cut-off, and the time occupied in a stroke.

The action that we have been considering is the principle obstacle to economy in the use of steam. In Section III. of this Part is shown the immense loss suffered from want of attention to it. The higher the pressure, the earlier the cut-off, and the better the vacuum, the greater will be the proportion of the entering steam condensed to restore to the surfaces the temperature which they must have before the steam will exert any dynamical energy. The combination of moderate superheating with the steam-jacket seems to afford the best means of avoiding this loss. Comparatively little benefit is derived from the jacket if the entering steam is wet. On the other hand, superheating in a degree sufficient to answer the purpose alone is unsatisfactory. Its degree cannot be precisely controlled, and the same degree may, if the steam is cut off early, be insufficient to restore the temperature to the surfaces, and yet sufficient to set the stuffing-boxes on fire if the steam is cut

off late. For slow-moving engines, superheating alone is of no account.

The waste room in the clearance and passage, or passages, at each end of the cylinder, is another source of loss, and one which cannot be got rid of. Generally it is necessary to add in this way about ·04 to the piston displacement. The proportion of loss from this cause increases as the steam is cut off earlier. Thus if we cut off at one-sixth of the stroke, the above amount of waste room adds one quarter, if we cut off at one-eight it adds one third, and if we cut off at one-twelfth it adds one half, to the weight of steam required to fill the piston displacement.

At the point of cut-off the steam contained in the waste room adds in the above proportion to the volume of the steam, but at the point of release it adds in nearly the same proportion to its density. It is not, therefore, wholly lost, but on the expansion adds its proportion to the force exerted.

This is illustrated on diagram No. 35, page 144. This, and No. 35*a* on the opposite page, were taken during the trial of the compound pumping-engine at Lowell, Massachusetts. The diagrams from the low-pressure cylinders, reduced to the same scale, are placed in their correct positions beneath those from the high-pressure cylinders. On diagram No. 35 the expansion, except for a short distance at the commencement, followed the Mariotte curve exactly. The amount of waste room is large, being ·06 of the piston displacement. The hyperbolic curve A, of which the diagonal *b* is the axis, represents the expansion of the steam contained in the space through which the piston has moved. The waste room, represented at the end of the diagram, adds ·4 to the weight of steam used, and the area between the two expansion curves represents the work done in the high-pressure cylinder by this addition. When expansion is carried farther in a second cylinder, the benefit is obtained of the increased density of the steam at the end of the first expansion; but when the exhaust from the high-pressure cylinder is opened, the waste room in the passage and at the end of the low-pressure cylinder has to be filled, and in this case this, and the condensation on entering it, together, occasion the loss at this point as the diagram shows, of one-fourth the pressure in the large cylinder. We can there see, that if seven-tenths of the loss in the latter waste room could have been avoided, the weight of

steam represented by the curve A would have done the same work in the large cylinder that in fact was done. Compounding

is subject to a large deduction from its theoretical gain on this account; but of this more hereafter.

The loss occasioned by the waste room in a single cylinder

can be found by the method illustrated below:—Let 100 lbs. pressure be cut off at one-sixth of the stroke, and let 24· repre-

sent the length of the stroke, and 1· represent the waste room which will then add one-fourth to the piston displacement up

to the point of cut-off. Then $24 \div 4 = 6$, and $25 \div 5 = 5$; and so we are, in reality, cutting off at one-fifth, instead of at one-sixth, of the length of the chamber.

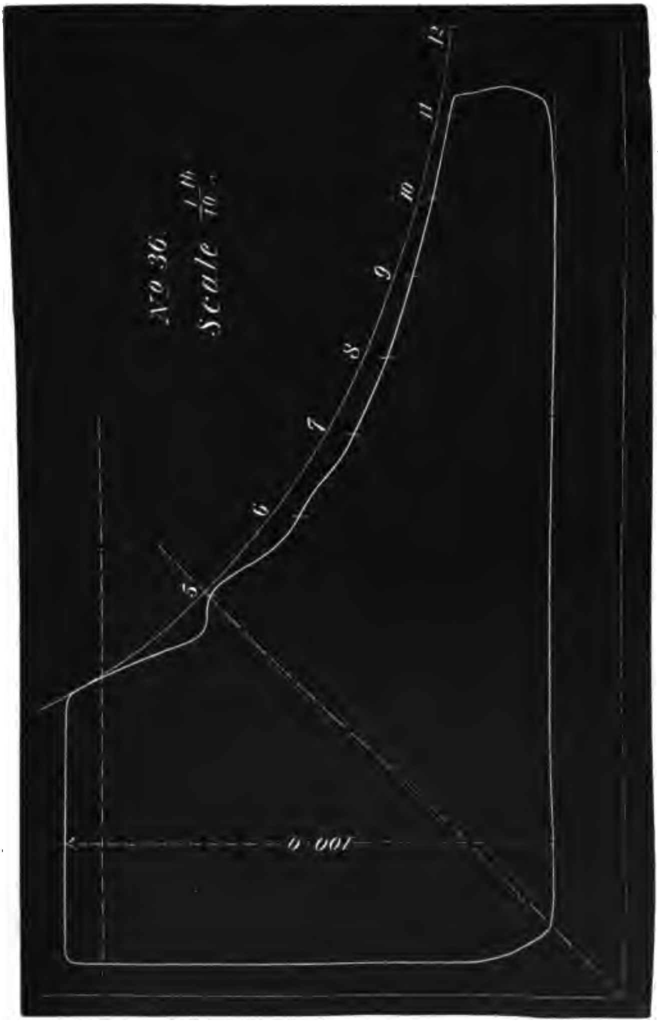
$100 \div 5 = 20 \cdot \times 2 \cdot 6 = 52 - \frac{1}{25}$ of $100 = 48$ lbs. mean pressure, and $100 \div 6 = 16 \cdot 66 \times 2 \cdot 8 = 46 \cdot 6$ lbs. mean pressure, during the stroke. $48 \div 5 = 9 \cdot 6$; and $46 \cdot 6 \div 4 = 11 \cdot 65$. The former number represents the work really done, and the latter that which, if there were no waste room, would be done by the same weight of steam. The loss shown is about 20 per cent. That is, 25 per cent. more steam is used, and about 5 per cent. is added by it to the work done. This loss is increased as we cut off earlier.

This loss can be completely avoided by compressing the exhaust steam up to the density of the steam in the chest. The steam filling the waste room then acts as a constant spring, giving out by its expansion the force needed to compress it again. The smaller the quantity of waste room, the higher will be the rise of pressure from the closing of the exhaust-valve at a given point of the stroke, and also the less the area of the cooling surface; so that the gain from reducing the waste room is threefold.

We have observed that the expansion curve in diagram No. 35 follows the law of Mariotte. If, however, the hyperbolic curve were drawn to coincide with the expansion curve at its commencement, then afterwards the latter would run all the way below it. The writer has analysed but one other diagram in which the expansion curve fell below the hyperbola all the way, and that is the following, No. 36, taken from a non-condensing Corliss engine at Saltaire; but it is to be apprehended that this was a case of a leaky exhaust-valve or piston. The waste room is very small, only ·033 of the piston displacement; the pressure was high, and steam followed a long distance; but there was no steam-jacket, and if the steam was not superheated the curve was impossible, while if it was superheated it would seem that the great heat of the cylinder should have raised the expansion curve higher at its termination.

In the very similar case of diagram No. 29, page 98, the valves and piston were known to be tight, and the steam was superheated about 30 degrees. The pressure fell a little below the hyperbolic curve during a considerable part of the expan-

sion, but rose to coincide with it before reaching the point of release, undoubtedly from the superheating of the expanding steam.



In the absence of positive proof, it seems doubtful whether the expansion curve, unaffected by leakage, can ever fall,

through its entire length, below the hyperbola. If not raised to it at its termination by the re-evaporation of moisture, it will be so, apparently, by the superheating of the steam. We shall see that in all ordinary cases it is raised above it, and sometimes far above it, by the former of these operations going on during the expansion.

In the next section we shall consider more fully the use of the Indicator in connection with this whole subject.

SECTION II.

OBSERVATIONS ON THE SEVERAL LINES OF THE DIAGRAM.

BEFORE considering the features of the engine and the action of the steam which are indicated by these lines, it is important to remind the engineer of some points which are often neglected.

The first of these is the mechanical condition of the engine. The engineer should *know* that the piston and valves are tight. Unless they in fact are so, the diagram will not tell what the engine would be doing if they were; and unless the engineer knows that they are so, he cannot conclude with certainty about the causes of the features exhibited. For example, diagram No. 35, page 144, shows an expansion curve following closely the curve of Mariotte, and diagram No. 35*a*, on the opposite page, taken simultaneously from the opposite end of the same cylinder, shows an expansion curve much higher at the termination of the stroke. The curve marked A is the Mariotte curve, drawn from the termination of the stroke. Why this difference?

The committee appointed to ascertain the duty of the engine came to the conclusion that the valve must have leaked. Subsequent observations of diagrams from other vertical engines lead to the belief that this conclusion was a mistaken one, and that the rise of pressure was due to re-evaporation; but the condition of the engine in this respect had not been ascertained, and so the answer to the above question was not then found.

Again, the capacity of the clearance and passages, which, as well as the piston displacement, are to be filled with steam, must be known, or neither the consumption of steam nor the character of the expansion curve can be determined.

The point of cut-off should also be precisely known. In the best expansion engines, running at moderate speed, this is

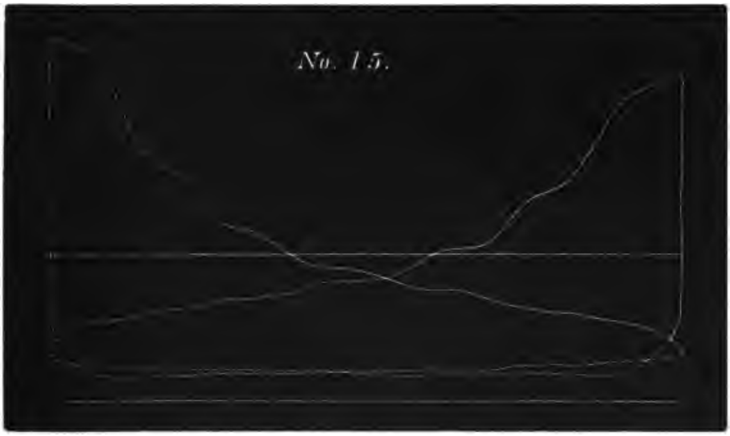
clearly indicated on the diagram; on diagrams No. 9, page 96, and No. 23, page 110, it would be hard to guess at it. On diagrams from engines running at high speed it cannot, as a general thing, be shown with precision, on account of the fall

of pressure in the cylinder during the closing of the port. There are several ways of determining this point when the diagram does not show it. If the cut-off is fixed the steam-

chest can be opened, and it can be seen; but the better and readier way is to take a diagram at slow speed, as in No. 19 in the Appendix. On locomotives the point of cut-off for each notch can thus be determined.

On engines with a variable cut-off this point can generally be known by the position of the governor, according to a scale made for the purpose. When, however, the point of cut-off cannot be determined, the hyperbolic curve with which its expansion curve is to be compared may be drawn, starting from a point on the expansion curve after it is certain that the steam has been cut off. When the expansion curve presents oscillations, the point of cut-off will be the highest point at which the hyperbola will touch it; and sometimes this will be at some distance from what looks like a sharp cut-off.

Of serrated and waving lines.—The characteristics of these lines will be considered together. There are none of more interest and value. They may occur at the points of admission, of cut-off, and of release—whenever the impulse received by the piston of the Indicator in either direction is so sudden that the momen-



tum imparted carries the pencil beyond the point which marks the real change of pressure. These are of great value, as they show precisely the degree of suddenness or violence of the action of the steam, and also the excellence or otherwise of the action of the Indicator. The old belief that they represented actual pulsations of the steam, and how quickly that was exploded by


No. 17

the application of this Indicator, have already been noticed on page 20.

The diagram No. 20, page 150, taken from the "Allen" en-

gine, running at the Paris Exposition of 1867 at 200 revolutions per minute, affords a beautiful illustration of this action. The pencil happened to touch the paper at the point A of the return

stroke. On the next return stroke the slight pressure prevented any vibrations, and the straight line was drawn through the mean of the previous ones. There was no fall of pressure at the release to produce any vibrations, and the vibrations which had taken place during the return stroke, when they were entirely unchecked, were the feeble termination of those which commenced at the point of cut-off, and which were then, of course, stronger than they show when checked by the friction of the pencil. The flowing character of the lines pleases the eye by the graceful undulations they present, and demonstrate the entire absence of friction in the cylinder of the Indicator.

Where at high speed the cut-off is sharp, the amplitude of the vibrations will be in a large measure controlled, especially if a weak spring is used, by the degree of pressure applied to the pencil. Very interesting lines are then obtained by letting the pencil run three or four times round, varying the pressure on each expansion from the least possible to a somewhat firm one.

Diagrams No. 15, page 151, and No. 17, page 152, exhibit, in contrast with the above, evidence of slight friction in the instrument; the fall is by a succession of steps, there is no rise of the pencil, no reaction. Neither of these was taken with the Richards Indicator. The former is from one of a pair of engines driving a screw at about 50 revolutions, and the latter is from a paddle-wheel engine making only 16 revolutions per minute. Of course, in both these cases the expansion curves should have been drawn without any vibration at all.

The diagram No. 28, page 153, from the cylinder of a gas-engine, affords at once a fine illustration of this vibration of the Indicator, and of its usefulness. It tells the whole story. When the stroke was two-thirds completed, and the excess of back pressure was beginning to stop the engine, the contents of the cylinder were exploded with a suddenness exactly represented by the extent and rapidity of the vibrations, and developed a force during the remainder of the stroke represented by their mean.

A slight vibration, producing long and gentle undulations on the return stroke, is shown in diagram No. 24, on the opposite page. This diagram was taken from Shaw's Hot-air Engine at the Paris Exposition of 1867, an engine which gave a promise that for some reason does not seem to have been ful-

filled. The vibrations were occasioned by the suddenness of the exhaust.

This use of the Indicator is of especial value in exhibiting the

character of the admission in high-speed engines. It is found that a very moderate pressure may be so admitted as to produce vibrations of the pencil of extreme violence; and on the other

hand, at the very same speed, the highest pressures may be obtained, precisely on the dead centre, so gently as to impart to the instrument scarcely any vibration at all.

All the Indicator diagrams in the book may be usefully studied in connection with the following remarks on each of the several lines, though only a few of them will be referred to.

I. The Admission Line.

This line is formed by the rise of pressure in the cylinder as the port is opened for the admission of the steam. It begins at the termination of the compression. In the old beam-engine days this was the point of anxiety. To admit the pressure gradually as the piston advanced, so as to save the beam centre from the strain, and the whole structure from the shock, of admission on or near the dead centre, was the constant aim, to which everything else was sacrificed. It is becoming difficult to realise the degree of this apprehension in the minds of the past generation of engineers. The writer was once assured by a high authority in England, that, if he saw any way to do it, he would not admit the steam till the piston had made a quarter of its stroke. Diagram No. 23, page 110, would present, in the eyes of our fathers, a very moderate inclination of the admission line; and they would have received from No. 12, page 85, as great a shock as any engine could do.

But time has been changing all that, and now, in direct-acting self-contained engines, we admit any pressure as plump on the centre as we can do; and find, moreover, that the faster we run the more complete is the absence of all shock or strain at that point.

The direction and height of the admission line, relatively to that showing boiler-pressure, are determined by the amount of lead given to the valve, for which no general rule can be laid down. It depends on the speed of the piston, the proportion between the area of the port and that of the cylinder, the rapidity or slowness of the opening movement, and the density of the steam already in the cylinder at the instant of opening. The correct lead can be found only by the application of the Indicator. Without its assistance the best judgment is likely to err, in a case presenting novel conditions. By the "best

judgment" is meant a judgment formed by careful comparison of the actual lead given, with the admission line drawn by the Indicator, in a wide diversity of cases.

II. The Steam Line.

This is drawn by the advance of the piston while the port continues open for the admission of steam to the cylinder. Here we find engines divided into four classes, namely:—

1. Those in which the valves have an invariable motion without any, or with only very trifling lap, causing the port to remain open, or, technically, the steam to follow the piston, quite or nearly to the end of the stroke.

2. Those in which the valves have also an invariable motion, but with more or less lap, causing the steam to be cut off at a certain fixed point of the stroke.

3. Those in which the point of cut-off may be varied by hand either by means of the link motion or of an independent cut-off gear; and,

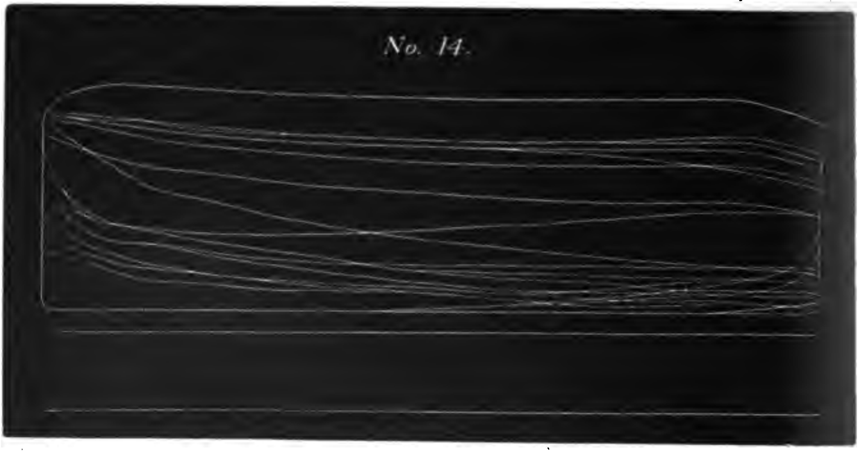
4. Those in which the point of cut-off is adjusted by the action of the governor, according to the changes either in the pressure of steam or in the resistance to be overcome.

In the first two classes, when less than the full pressure is required in the cylinder, the governor or the engine-driver adjusts the pressure by changing the position of the regulating valve. In the third class the regulating valve may be employed for the same purpose, but the more usual and better way is, to run such engines with this valve entirely open, and to adjust the mean pressure in the cylinder by changing the point of cut-off. Engines of the fourth class have no regulating valve, but the full attainable pressure of steam is admitted to the cylinder.

The action of the regulating valve varies the *position* of the steam line upward or downward, to that distance from the atmospheric line which gives the mean pressure required. The action of the cut-off gear, on the contrary, varies its *length* for the same purpose. In engines in which the steam follows to the end, or nearly to the end, of the stroke, and indeed in all cases where the pressure is reduced between the boiler and cylinder by the action of the regulating valve—that is, by throttling—it is a matter of very little interest what the steam line may be. Not only its distance from the atmospheric line, but also its direction,

is changed by every change in the position of the regulating valve, so that it is not at all a fit subject for consideration, as is well illustrated in diagram No. 14, taken from a non-condensing engine, where, owing to the bad action of the governor, the position of the regulating valve varied extremely at every stroke of the engine. The diagram shows the steam lines for fifteen consecutive revolutions, and illustrates the value of a good fly-wheel. The two lines drawn under the diagram are the atmospheric line and the line of perfect vacuum. These lines are essential to a proper knowledge of the quantity of steam consumed.

In engines which have no regulating valve, or where it is



not employed, as in marine engines except in rough weather, the steam line should approach nearly to the line of boiler pressure, and should be parallel with this line up to the point of release or cut-off. As, however, a difference of level is necessary to give motion to a stream of water, so inequality of pressure is required to produce a current of steam, the amount of the inequality depending on the velocity of the current, or on the area of the passage where most contracted, and on the number and amount of its deflections. For this reason neither the pressure in the boiler, nor the vacuum maintained in the condenser, can ever be reached in the cylinder, although the employment of large ports and pipes free from bends will

enable both to be nearly approximated to. Diagrams Nos. 1, 10, 11, and 17, afford examples of correct steam lines, except that in No. 1 it is not continued parallel nearly up to the point of cut-off. Diagram No. 12 shows a slight fall of the steam line as the piston advanced, but the point of cut-off is well shown. Diagram No. 15, from a marine condensing engine, at 336 feet travel of piston per minute, and Nos. 4, 5, 6, and 7, from a locomotive, at 730, 820, and 950 feet travel of piston per minute, afford, on the contrary, examples of bad steam lines. The boiler pressure is very nearly attained at the commencement of the stroke, in the first case by lead given to the valve, and in the others by lead superadded to excessive compression; but as the piston advances, the pressure falls with great rapidity, and the point at which the port was closed the diagram affords no means of discovering. In all these cases the passage of steam to the valve-chamber was entirely unimpeded.

The nature of the steam line depends principally on the proportion between the area of the ports, supposing them to be, as they ought, the smallest passages through which the steam is taken, and the capacity of cylinder to be filled in a given time. A given capacity may be formed in the same time by the slow advance of the piston in a larger cylinder, or by its more rapid advance in a smaller one. The sectional area of cylinder and the speed of the piston must, then, be equally considered in determining the area of the ports, as they are equal elements in determining the capacity of cylinder to be filled. The velocity of the current of steam is really the only point to be considered. This, through the port, should not exceed 200 feet per second.

While, therefore, very high velocity of piston does not render impossible the attainment of a correct steam line, still the size of port required for this purpose becomes so considerable, and the amount of power absorbed in working the valves, under the pressure which is generally associated with high speed of piston, is already so serious, that with the present form of valve in use—on locomotives, for example—it is better probably to submit to the defect at high velocities, than to attempt to amend it by enlargement. Improvement in this feature can be looked for only from a radical change in the valves and their movements.

The following diagram, No. 37, has been taken at speeds increasing from about 40 revolutions to 170 revolutions per minute, expressly to show how, all other circumstances being

the same, the steam line falls as the speed increases, through the insufficiency of the port. The point of cut-off was fixed; the vacuum was being formed while this diagram was taken. It illustrates also the lengthening of the diagram that is caused by the elasticity of a long cord at high speed.

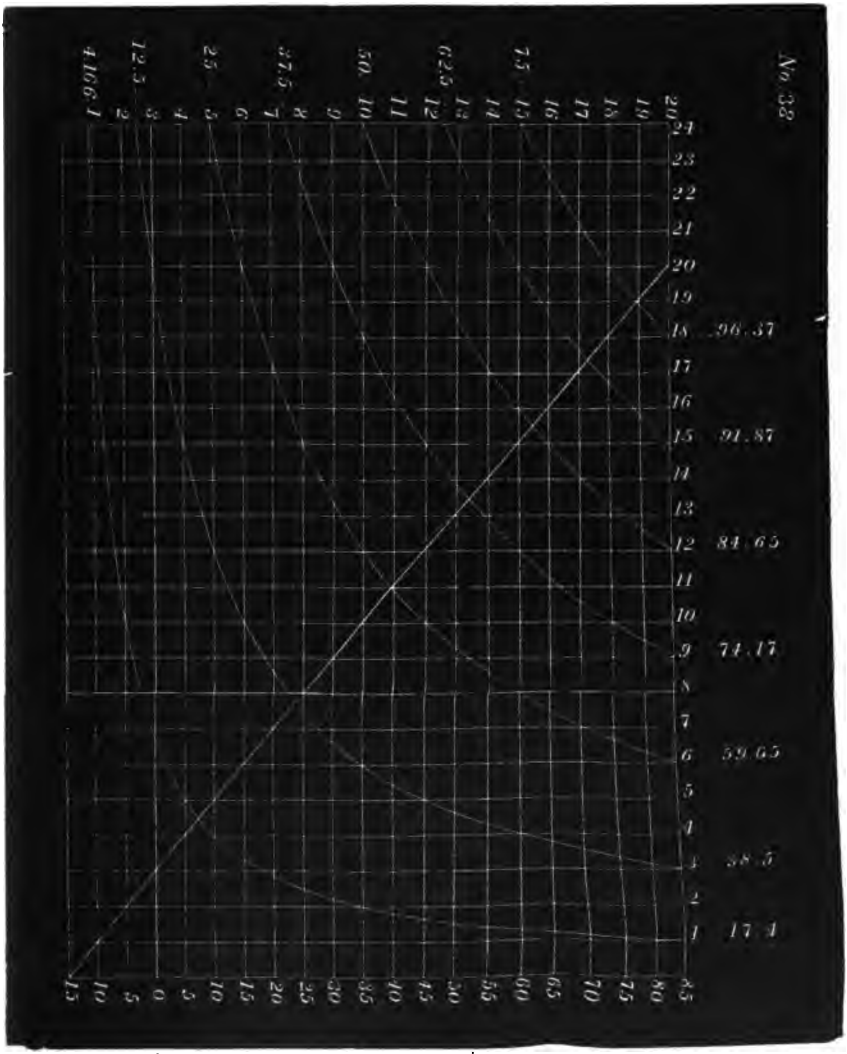
Another cause also contributes to injure the steam line, especially in condensing engines, namely, the condensation of the steam on entering the cylinder. To this cause the great fall of pressure in diagram No. 15 must in part be attributed, the small size of the ports not being sufficient to account for it, at the speed of piston employed, of 336 feet per minute.

III. The Expansion Curve.

If any one line of the diagram possesses a greater interest than the others, it is the expansion curve. We have already become familiar with it, and have found it a theme of never-ending interest.

The Mariotte curve, as has already been said, is the standard by which the character of all expansion curves actually drawn is determined. This it is for various reasons. It is a determinate mathematical curve, a hyperbola, it can be readily and

precisely drawn, and the best curves attainable in practice coincide with it very nearly, if not exactly. We will attend first,



then, to the proper method of drawing this curve, and of applying it to any diagram.

The preceding figure, No. 38, has been drawn to illustrate the application of this curve to the expansion of steam. The pressure represented by the height of the parallelogram is 100 lbs. on the square inch, and its length represents the stroke of a piston, including the waste room, divided into 24 equal parts. The base represents the absolute vacuum, and the right-hand boundary the commencement of the stroke.

The diagonal of a square, drawn from the point of intersection of these lines, is the axis of every hyperbola that can be described, representing expansion, according to the law of the gases, from any point of the stroke, and from any pressure whatever.

Curves are described representing this expansion from seven different points of cut-off. The figures at the terminations of the curves give the terminal pressures, and those at the commencement give the mean pressures during the stroke, as follows:—

Proportion of the stroke at which the steam is cut off.	Terminal pressure representing the steam consumed, in lbs.	Mean pressure, representing the power exerted, in lbs.	Proportion of power exerted to steam consumed.	Absolute gain; 1· representing the power exerted by the same steam without expansion.
(a)	(b)	(c)	(d)	(e)
$\frac{3}{4}$	75·	96·37	1·285	·285
$\frac{15}{24}$	62·5	91·87	1·469	·469
$\frac{1}{2}$	50·0	84·65	1·693	·693
$\frac{9}{24}$	37·5	74·17	1·978	·978
$\frac{1}{4}$	25·0	59·65	2·386	1·386
$\frac{1}{8}$	12·5	38·5	3·080	2·080
$\frac{1}{24}$	4·166	17·4	4·177	3·177

The theoretical possibility of gain by expansion is thus illustrated. The various practical limitations, and the degrees in which these may be avoided, have been already considered. With respect to the principal cause of loss, namely, the condensation of the entering steam, the means are now well understood by which this may not only be entirely prevented, but

may be, and is, changed into a gain by the superheating of the steam during the expansion.

It is interesting to observe throughout this diagram, the gain that is realised by expanding from higher pressures. In every case, all the work shown above any abscissa is clear gain from the higher pressure. For example, as will be seen, 100 lbs.

cut off at $\frac{6}{24}$ths shows a large gain over 60 lbs. cut off at $\frac{10}{24}$ths, and again, though not so large, over 75 lbs. cut off at $\frac{8}{24}$ths of the stroke; 100 lbs. cut off at $\frac{1}{24}$th gives, from the same steam consumed, very nearly twice the power that is given by 20 lbs. cut off at $\frac{5}{24}$ths of the stroke. Column (*e*) of the preceding

Table gives the powers represented by the areas included between the expansion curves and the abscisses drawn to their points of termination.

To apply the hyperbola to a diagram.—Draw the line of absolute vacuum, and perpendicular to it the line representing the addition made by the waste room to the piston displacement,

and from their point of intersection draw the diagonal of a square. This is the axis of the curve. It gives the proper

conception of the character of the curve, and aids in its construction and verification, since corresponding portions on either side of it are alike.

The hyperbola may be commenced at either end of the expansion curve. Diagram No. 21, page 103, illustrates one method, and diagram No. 26, illustrates the other. Generally it will

be found the more accurate way to commence near the point of release. A point having been selected, divide the length of the diagram up to this point, including the waste room, into any convenient number of equal parts, the more numerous the better.

Multiply the pressure at the point selected into the number of parts, and the product will be the pressure, according to the law of the gases, at the end of the first of these parts, supposing it to be carried so high; and the pressure at the end of any part is found by dividing the pressure at the end of part one by the number of parts up to such point.

One of the above pair of diagrams, No. 40, from a Corliss engine, has been divided to illustrate this method. At the end of the thirteenth division the pressure is 9 lbs., so at the end of the first division it would be 117 lbs.; at the end of the second one, 58½ lbs.; at the end of the fourth, 29¼ lbs.; at the end of the sixth, 19½ lbs., and so on. The method is extremely simple, only care must be taken that ordinates, and not diagonal lines, are measured. Above the axis the points of intersection of the ordinates with the curve become separated too far for accuracy, as seen in diagram No. 22, page 105. It is then better to substitute the abscisses as shown in No. 40. These are numbered to correspond with the ordinates, and then the lengths are transferred from the latter. The curve passing through these points is the hyperbolic curve.

It is evident that, without making application of the hyperbola to an expansion curve, we can know nothing at all about it; but that as soon as this has been done its features all stand revealed, and the causes by which these have been produced are either at once suggested, or we are put on inquiry to discover them. Diagram No. 40, for example, shows condensation continuing after the steam had been cut off, and passing slowly into re-evaporation. The commencement and progress of the latter action is seen in the expansion-curve ceasing to depart from, and then approaching, and finally crossing, the hyperbola. The loss thus shown here, however, is not nearly so great as in diagram No. 22, taken when the steam was cut off still earlier.

It is certain that this application has not heretofore been made on one diagram in ten thousand; indeed it is very rarely that the waste room in a cylinder is known, even by the designer or the builder of the engine, so that the curve can be drawn. It is to be hoped that its importance will come to be more generally understood.

The preceding diagrams, Nos. 25 and 26, were taken from the engines of two gun-boats of the American Navy, the *Winooski* and the *Algonquin*, during the civil war, on a trial of rival systems, which at the time attracted a good deal of attention.

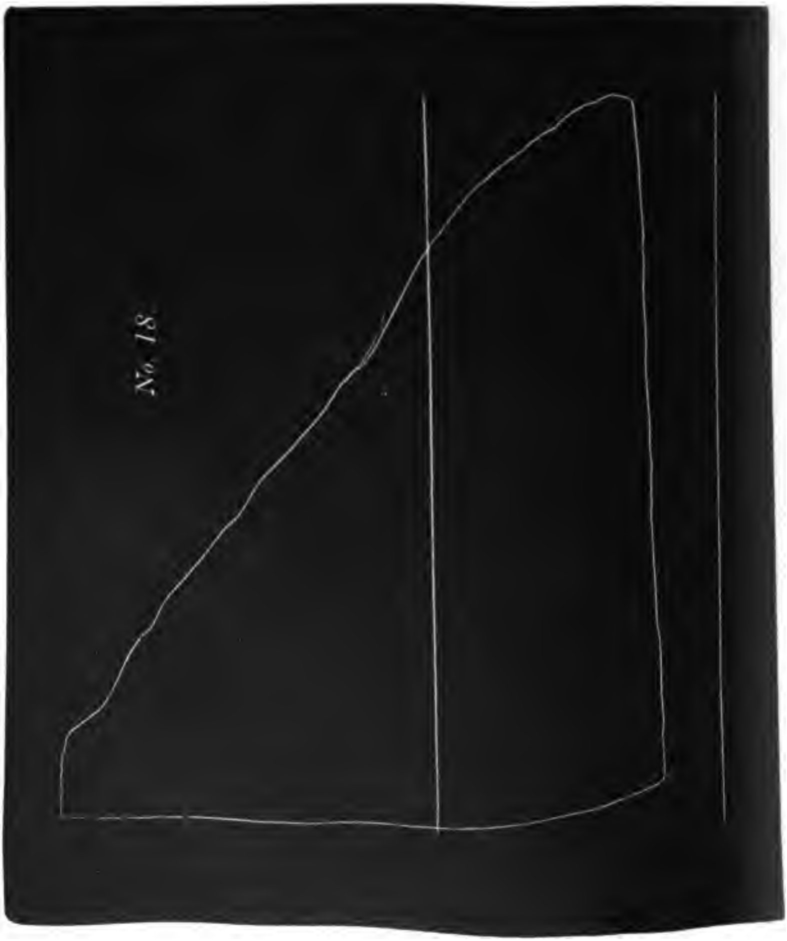
The former were designed by Mr. Isherwood, then Engineer-in-chief of the Navy, and the latter by a Mr. Dickerson, to demonstrate the superiority of the system of high pressure and expansion. The dotted curves show the correct expansion.

The truth found a bad advocate. The diagrams show that the admission and the cut-off engrossed the attention of the designer of the latter engines to the exclusion of everything else. These were perfect, but the release was absurdly early and slow, and the vacuum averaged only 9 lbs. The necessity, not only of keeping the cylinder hot, but even of preventing water from coming over from the boiler, seems never to have been thought of. The cylinders were of about equal size, the mean pressures nearly alike, and the terminal pressure, which in the *Algonquin* should not have been one half that in the *Winooski*, was nearly the same in each. The release and the vacuum in the latter were simply magnificent. The cause of high pressure and early cut-off met with overwhelming discomfiture, these being transformed into means of loss instead of gain. If, however, the diagram measured at the termination of the expansion curve showed the quantity of heat used, the *Algonquin*, in spite of its enormous re-evaporation, should still have done a very little the better. This was, however, only an indication of the far greater re-evaporation which took place during the return stroke. The explanation of its failure is found in this. The boilers were very small. When boilers are disposed to prime, the pulsations of the steam produced by cutting off early cause this action to go on in an excessive degree. This accounts at the same time for the great quantity of feed-water that had to be supplied to the boilers of the *Algonquin*, the almost inconceivable rise of the expansion curves, the very poor vacuum, the condenser being filled with hot water, and the excessive quantity of coal consumed. No condensation and re-evaporation of dry or tolerably dry steam, at the speed at which these engines were run, would begin to account for any of them. In the diagram No. 26 two things are to be noted—first, re-evaporation commenced as soon as the steam had been cut off, which seems never to be the case unless a great deal of water is brought over; and second, the re-evaporation was much greater in that end of the cylinder in which the steam was cut off earliest. It might naturally be supposed that the one-idea people would have learned a lesson from this trial—that if economy is to be assured, a good many other things need to be attended to besides

cutting off early and sharp; but they do not appear to have done so.

We can now understand also why diagrams like No. 27, although very pretty to look at, are very wasteful to produce. We see here, even without drawing the theoretical curve, the

condensation continuing during the early part of the expansion, and the large amount of re-evaporation that followed it; the terminal pressure, though below the atmosphere, being about twice what it ought to be.



It is to be observed, that the expansion should in this case fall just as low as though the engine were a condensing engine; since, in either case alike, this is the enlargement of volume of the steam in a closed cylinder.

It is not likely that large and costly engines will be made much longer, to make such diagrams as these, developing very little power, under the delusion that all this is necessary for economy.

The practical limitations to the benefit to be derived from expansion are coming to be understood; and economy will be more successfully sought in moderate degrees of expansion, with proper regard also to other conditions on which it is dependent.

To expand steam properly, it is doubtless necessary that it should be cut off with reasonable quickness—that the pressure should fall very little during the closing of the port. Diagram No. 16, page 163, for example, shows a fair compliance with this requirement—quite sufficient for all practical advantage. Sharper cut-off than this is pretty, but the gain from it is not appreciable. On the other hand, the excessive wire-drawing shown, for example, in No. 1, page 12, in No. 18, opposite, and in No. 23, page 110, is largely destructive of gain by expansion. The expansion never properly begins till the port is closed; as for example, at the dots in No. 16. On diagram No. 1 an expansion curve has been drawn to show how one-half of the steam, cut off at three-eighths of the stroke, would have done three-quarters of the work.

Compound cylinder-engines.—The consideration of the subject of expansion would be quite incomplete without a reference to compound engines. The Wolfe system has had a remarkable career. It was originally introduced when very low pressures were employed, and expansion was little known, and that which was used was what could be obtained by adding lap to the valve. With it was commenced the working of steam at higher pressures. Of course it showed a large economy. About the year 1860 it was applied, by the late Mr. Humphreys, on the ships of the Peninsular and Oriental line, in connection with superheating, the steam-jacket and surface condensation, and for a considerable time was regarded as the principal element of an economy before unknown in marine engineering. Its apparent success led to its revival on stationary engines, especially in Lancashire and Yorkshire. Six or seven years more saw the manufacture of compound engines pretty much abandoned, both for sea and land purposes. Now it again challenges criticism, its use having within the past six years become nearly universal in ocean steamers, which are run with a degree of economy which, though very unequal, sometimes

reaches to that which first in the *Mooltan* astonished the engineering world.

It is to be observed that compounding is never tried by itself, so as to stand or fall on its own merits; but is always associated with some great and unquestioned economic improvement. This time, singularly repeating its first introduction, it has been revived in connection with perhaps the most marked advance ever made in marine engineering; namely, raising the steam-pressure, almost at one jump, from about 20 lbs. to from 60 lbs. to 75 lbs. on the square inch above the atmosphere. Less attention seems to have been paid to the increase of pressure as an element of economy than it deserves. Our previous examination of the subject has shown its great value in this respect.

This is only the application of the system of Mr. McNaught, by which, years ago, he so admirably pieced out the power of the old cotton-mill engines, providing a higher pressure of steam, and also a cylinder to work it in before it should enter the old cylinder at the old pressure. The fact is, the McNaught idea—a happy makeshift, planned to furnish the increased power required by the enlargement of the cotton-mills, and at the same time utilise their old enormous and feeble beam-engines, and avoid putting any additional strain on their beam centres—proving a marked success, took complete possession of the Scotch engineers; so that to this day the two ideas, of a high pressure of steam and two cylinders to work it in, are inseparably associated in their minds; and the American marine engineers only copy the Scotch, and so are innocent of any idea at all on the subject.

Compounding is merely a method of prolonging the expansion—that is all. It is claimed to be of value, also, as a means of equalising the pressure on the crank during its revolution. The questions of the distribution of rotative force through the revolution, and of relieving the crank from the impact of the steam on the centres, have, however, to be newly examined, in the light of the demonstrations presented in Part Fifth of this Treatise. At present we confine ourselves to a consideration of the economy of the compound system.

The question presented is: Supposing the same pressure, superheating, steam-jacket, and other recognised means of economy, to be employed in both cases, which plan is more economical—to expand to the same point in one cylinder, or in two? The above reference will explain why, so far as relates

to ocean navigation, the question has not been settled experimentally, and why, when the importance has been felt of getting rid of the intermediate chamber, and having an equal force applied to each crank, engineers have even made four cylinders, setting a little one on top of each big one, rather than complete each expansion in a single cylinder.

The diagram, however, helps us to answer this question. Let us look first at the diagrams from the steamship *Egypt*, pp. 96, 97. We observe the only partial admission of the steam-pressure—6 lbs. short at one end, and 10 lbs. at the other—the fall of the steam-line, the late cut-off, nearly at the half-stroke, the short expansion and early release in the small cylinder, and in the large one the poor vacuum—all features prejudicial to economy, but none of them pertinent to the present discussion.

In order that the expansion which is begun in the small cylinder shall be continued without interruption in the large one, two things are required. The first is, that the forward pressure in the latter cylinder shall be equal to the back pressure in the former one. This depends entirely on the area of the communication. No matter how much of its pressure may be lost by the steam in passing from the high-pressure to the low-pressure cylinder, or from what cause; if it is not throttled between them, there will be an equilibrium of pressure on the two pistons.

The second requirement is, that there be no fall of pressure between the two cylinders. But this is impossible, says the compound engineer, and you cannot require what is impossible! The amount of this fall is what is mainly to determine whether or not we shall condemn the system: continuous expansion requires that there be no such break at all. In this case the capacities of the two cylinders are to each other as 1 to 3·5*

The dotted lines on these diagrams show the pressures that would be exerted if the above requirements could be complied with. We observe that there was a fall of pressure between the two cylinders of from 9 lbs. to 9½ lbs. Let us consider for a moment what this is. It is a sudden loss of 30 per cent. of the force of the steam, in the very middle, or rather in the earlier

* It would be a good practice always to indicate the cylinders of compound engines on scales differing inversely as their capacities, as, for example, in this case 35 and 10 lbs. to the inch, when the diagrams would represent the respective powers exerted in each cylinder.

part, of the expansion. This great fault—to use a geological expression—in the expression curve, is found in all diagrams from compound engines, and this example is believed to show a fair average of its amount in marine engines.

But now we find a remarkable feature presented. The mean back pressure in the high-pressure cylinder is 9 lbs. higher than it ought to be from a terminal pressure 9 lbs. higher, occasioning a loss of 375 horse-powers in this cylinder. In the low-pressure cylinder, although the steam is cut off at the half-stroke, the pressure falls before the release but little, if any, below that to which it should fall by continuous expansion from the initial pressure 9 lbs., or 40 per cent. higher, and rises so high before it is cut off as to make up nearly one-fourth of the loss above noted in the high-pressure cylinder.

This is the measure of the re-evaporation in the intermediate chamber, and in the low-pressure cylinder, produced by the heat imparted from the jacket. We see this begun in the high-pressure cylinder, the curves A B representing the correct expansion. This, besides the direct loss of 10 to 12 per cent. on the power of 2800 horse exerted by these engines, shows a great waste of heat; for, in order that re-evaporation may take place, there must first have been at least an equal condensation. This feature will be found commonly shown in diagrams from compound engines.

In diagrams Nos. 35 and 35*a*, pp. 144 and 145, the expansion is shown carried to a pretty low point in the first cylinder. The continuation of it in the second cylinder exhibits a large re-evaporation, preceded by a fall of pressure of 3 lbs. at one end, and 5 lbs. at the other. It is curious to observe, that the lesser fall of pressure takes place in passing from that end of the first cylinder in which the re-evaporation had been much the greater. We see this greater re-evaporation continued thus while the pistons are moving slowly near the dead points of the crank, and then through the expansion in the low-pressure cylinder.

These very different illustrations show the chief use of the second cylinder to consist in re-evaporating, by means of the jacket, water that ought never to have been formed.

Undoubtedly, the compound-cylinder engines show, on the average, a good degree of economy; and undoubtedly, too, it is expanding from a good pressure and to a low point, with the use of the steam-jacket and superheating, to which, in spite of

the loss involved in the use of two cylinders, it is all due; and the same expansion in independent cylinders, under the same conditions, will show a considerably better result. Take, for example, diagram No. 12, page 78, in which a total pressure of 35 lbs. is expanded down to a little less than 7 lbs. Suppose a total pressure of 85 lbs. to be cut off proportionately earlier, so that the added 50 lbs. is represented by an addition above the present diagram, as has been already explained; and suppose the cylinder to be warmed by the steam-jacket and the steam to be superheated, so that not only does the entering steam suffer no condensation, but its expansive force is increased during the expansion, as is the case in engines in which compounding is successful; then, a smaller cylinder being used on account of the addition to the mean pressure, and there being no fault in the expansion curve, who can doubt that a better economy would be attained than by compounding with all its complications?

It is often remarked, "But when you cut off so very early you must have a larger cylinder." The contrary is the fact, and that in an enormous degree, as may be seen in an instant. Take the second curve in diagram No. 38, page 161. Let one twenty-fourth of the parallelogram represent the waste room, then the curve would represent 85 lbs. pressure above the atmosphere, cut off at about one-twelfth of the stroke. Now it has until recently been a common practice in marine engines to cut off 20 lbs. above the atmosphere at about one-third of the stroke. If the reader will look along the abscissa numbered 20, he will see it strike this expansion curve at about this point of the stroke, previous to which time all the work represented by the area above it has been done by the higher pressure; and on measurement, omitting the waste room, and allowing 2½ lbs. back pressure, this will be found to add 50 per cent. to the work done, the addition being one-third of the whole. It will be observed that elevation of the pressure adds power at one end of the stroke, precisely as prolongation of the expansion adds power at the other end. But the size of cylinders will be still further reduced, and the economy improved still more, by carrying this pressure to a later point of cut-off,—to one-eighth or one-sixth of the stroke.

It is claimed that the use of two cylinders avoids the exposure of the same surface to the extremes of temperature on the opposite strokes. But these extremes of temperature are avoided

only in a slight degree, sometimes scarcely appreciable, as seen in diagram 35, while the extent of surface to which the same steam is exposed is greatly increased. It is not exposure to cooler steam but to cooler water that is to be dreaded. But nothing is thought even of immersing the high-pressure cylinder in a bath 100° cooler than the entering steam, if only this can be worked through two cylinders. The improved economy shown by compound marine engines is by comparison with the engines previously used in similar service, which were unjacketed, and in which steam of low pressure was worked, and the economy is not always better than it was in those.

Compounding has just won a great victory in America. A compound-cylinder engine, *both cylinders jacketed*, has beaten single-cylinder engines, *unjacketed*, evaporating so little as 18 lbs. of water per horse-power per hour; and the details of the triumph are in all the papers.

IV. The Exhaust Line and Line of Counter-Pressure.

These two lines may properly be considered together. The problem is, first, to employ the full expansive force of the steam as nearly as practicable to the end of the stroke, and then immediately to discharge, as far as possible, the pressure that will hinder the return of the piston. So far as the mechanism of the engine is concerned, this is effetced, first, by opening an exhaust-port of sufficient area, and opening it very rapidly; and, secondly, by providing a passage to the atmosphere, or to the condenser, much larger than the area of the port, so that the steam, after having passed the latter point, shall meet no obstruction at all. In a non-condensing engine nothing further than this is required. In a condensing engine the degree in which the back pressure can be removed depends, of course, upon the tenuity of the artificial atmosphere maintained in the condenser.

The diagrams here presented exhibit a variety of these lines, both in condensing and in non-condensing engines. Their careful study is recommended. Some present a remarkable conformity with theoretical requirements, and others show quite the reverse. There has undoubtedly been a great improvement in engines generally in this respect within the twelve years since the introduction of this Indicator. The various faults, of opening the exhaust too early and too late,

too slow and too little, have received more attention, as their consequences have been more observed on the diagram. In this, as in every other particular, we can truly say, that, so far as respects the distribution of the steam, we know nothing about the proper construction of an engine, or of its requirements, except what the Indicator has told us.

V. THE COMPRESSION LINE.

This line, when it exists, is formed by the closing of the exhaust-port at some point before the termination of the return stroke, when the advancing piston compresses the confined steam to a greater density. For the proper reading of this line, also, a knowledge of the extent of the waste room is indispensable. We have here piston motion converted into heat, the equivalent of the motion arrested, so far as this is done by the resistance of the steam. The heat thus supplied is mostly required by the rise of pressure of the steam. The total heat of one pound of steam must, for example, be increased in raising its pressure from 15 lbs. on the square inch to 30 lbs. 11·35 thermal units; to 45 lbs. 18·7 thermal units; and to 60 lbs. 24·26 thermal units.

The capacity of the remainder of the piston displacement and the waste room gives the volume of steam at the point of closing the port; and, its pressure being known, its weight is found by the Table. The force with which the piston compresses the steam, or the work—motion against resistance—lost by the piston, and imparted as heat to the steam, is shown on the diagram.

For example, in diagram No. 21, page 103, we have thus confined, as already ascertained (page 104), ·01482 of a pound of steam. Its pressure was raised by the compression from 16 lbs. to 45 lbs. on the square inch. The quantity of heat required to be imparted to it was $17 \cdot 7 \times \cdot 01482 = \cdot 262$ of a thermal unit.

The work done in compressing this steam was a mean of $15 \times 200 = 3000$ lbs. through nearly 1 inch, say 230 foot-pounds. This is the energy that there ceased, and appeared as ·3 of a unit of heat.

We may say, then, generally, that the piston, in losing its motion, supplies to the steam compressed the heat required by its increase of density, and something more. The compression

curve should therefore, if none of the heat were lost by the steam, rise somewhat more rapidly than the curve of relative volumes, which itself rises more rapidly than the Mariotte curve. (*See* diagram 34, page 131.)

It is observed in practice that the more frequent the strokes of the piston, and the drier the steam, the higher the compression curve rises. Often, after a rapid rise for a certain distance, a short curve in the horizontal direction marks the limit of density to which the steam could be compressed; because the water present absorbed the heat above a certain temperature, which was supplied by the arrest of the piston. Sometimes, owing to the rapid abstraction of heat from the steam, this curve turns downward, or there is a fall of pressure directly on the centre, if the admission is a trifle late.

Diagram No. 1, page 12, shows a rise of the compression line, from the low pressure of 2·5 lbs., which is much too rapid to be accounted for, except by a considerable leakage of steam from the chest, especially at the slow motion of the piston, making only fifteen double strokes per minute.

The link motion, when operating a single valve, varies the compression, as also the release, with every change of the point of cut-off. The excessive compression which the link motion effects when cutting off early, is one of its most valuable features, although an incidental one. It reduces the power exerted at both ends of the stroke, saves the steam contained in the waste room, and gives to the locomotive its smoothness of running, while its reciprocating parts are kept as light as possible. Diagram No. 19, in the Appendix, exhibits this action very well.

It would occupy too much space to enter upon a more detailed investigation of this interesting curve. Sufficient has been said to merely indicate the method of its proper analysis, which in many cases will be found to be full of interest and instruction.

SECTION III.

OF THE LOSS ATTENDING THE EMPLOYMENT OF SLOW-PISTON SPEED, AND THE EXTENT TO WHICH THIS IS SHOWN BY THE INDICATOR.

In the year 1860 a Board of Engineers, of which Mr. Isherwood was at the head, was appointed by the United States Navy Department to conduct a series of experiments, "for the purpose of determining the relative economy of using steam with different measures of expansion." These experiments were conducted with one of the engines of the U.S. paddle-wheel steamer *Michigan*, on Lake Erie.

The cylinder of this engine was 3 feet in diameter by 8 feet stroke. It was entirely unjacketed; but, as well as the boilers and steam-pipe, was protected against loss of heat externally. The waste room at one end of the cylinder added ·058 to the piston displacement. The temperature of the steam was not taken; but it must have been superheated in some degree, since it was first conducted into a large chamber above the boilers, but separate from them, through which passed the uptake from the furnaces.

The experiments revealed a loss by condensation of the steam on entering the cylinder, which increased as the steam was cut off earlier, and which is characterised in the Report as "the great antagonistic cause, that neutralises and reverses the economy promised by the theory of expansion."

The following figures, taken from this Report, give the data pertinent to the present discussion:—

Points in the Stroke at which the Steam was cut off.

	$\frac{11}{12}$	$\frac{7}{10}$	$\frac{4}{9}$	$\frac{3}{10}$	$\frac{1}{4}$	$\frac{1}{6}$	$\frac{4}{45}$
No. of revolutions made per minute	20·6	15·56	17·28	18·69	13·87	11·17	14·1
Pounds of water evaporated per hour, for each indicated horse-power exerted	39·94	34·8	33·08	35·2	34·47	37·04	46·08
Percentage of feed-water not accounted for by the Indicator	10·71	15·34	27·18	41·66	39·6	42·11	45·1

The following pair of diagrams, No. 50, are copies of those given in the official report of this trial, as taken when cutting off at the earliest point, or $\frac{4}{45}$ths of the stroke, with the lines

added representing the commencement of the waste room and the absolute vacuum, and the curve of expansion according to the law of Mariotte. The latter is designated on each diagram by the letters A B.

On comparing the actual expansion curve with this, we observe that the condensation of the entering steam did not cease when the flow of steam into the cylinder ceased, but continued after this had been cut off, and until the pressure had, by expansion, fallen below the atmosphere. The curve then begins to approach, and soon crosses the theoretical line, showing condensation passing gradually into re-evaporation. The cylinder was so cool, that, although the boiling point fell to about 160°, only a small portion of the steam condensed was re-evaporated during the stroke; but 45 per cent. of the water evaporated by the boilers existed in the state of water at its termination.

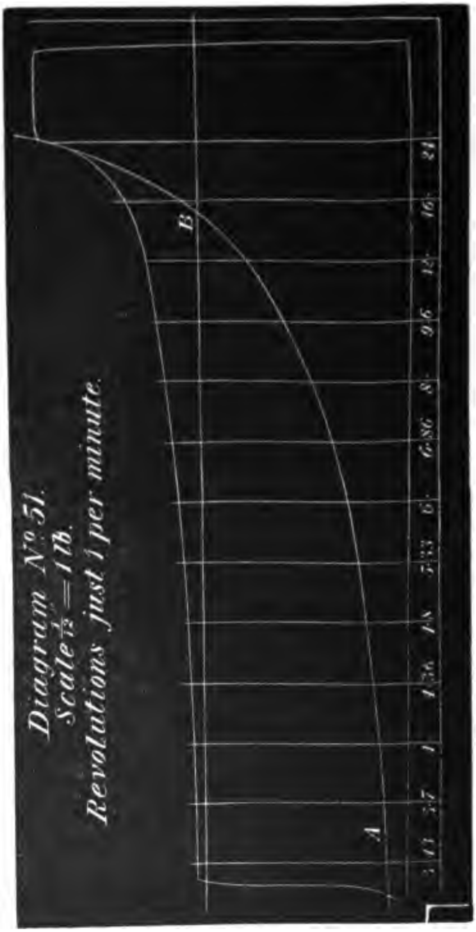
The presumption was pretty strong, that the enormous loss shown was in a large degree due to the slow speed of piston employed, and, in the light that has recently been thrown on the subject, this is now clearly seen to be the case.

A system of pumping machinery has been erected for the City of Providence, R.I., consisting of five steam-cylinders of 20 inches diameter and 36 inches stroke, and five water-cylinders of 12 inches diameter and 36 inches stroke, horizontal and double-acting. The steam-cylinders are of the Corliss pattern. The pressure carried is 60 lbs., and the steam is believed to be superheated 50° or more, as it is made in a vertical boiler, the tubes of which pass for 3 feet through the steam space, and this degree of superheat has been found in similar boilers elsewhere. The point of cut-off is fixed, and the pressure is reduced by throttling in the pipe. These engines are run at very slow speeds. As in the case of the *Michigan* the cylinders were unjacketed, but the external protection was complete.

The following five diagrams, Nos. 51 to 55, were taken by the writer from these steam-cylinders, at the speeds respectively of 1, 6, 10, 15, and 20 revolutions per minute. The cylinders were all indicated at each end, and diagrams similar to these were everywhere obtained. To these diagrams also have been added the lines representing the commencement of the waste room and the absolute vacuum, and the theoretical expansion curve, which, as in the preceding diagram, is designated by the letters A B.

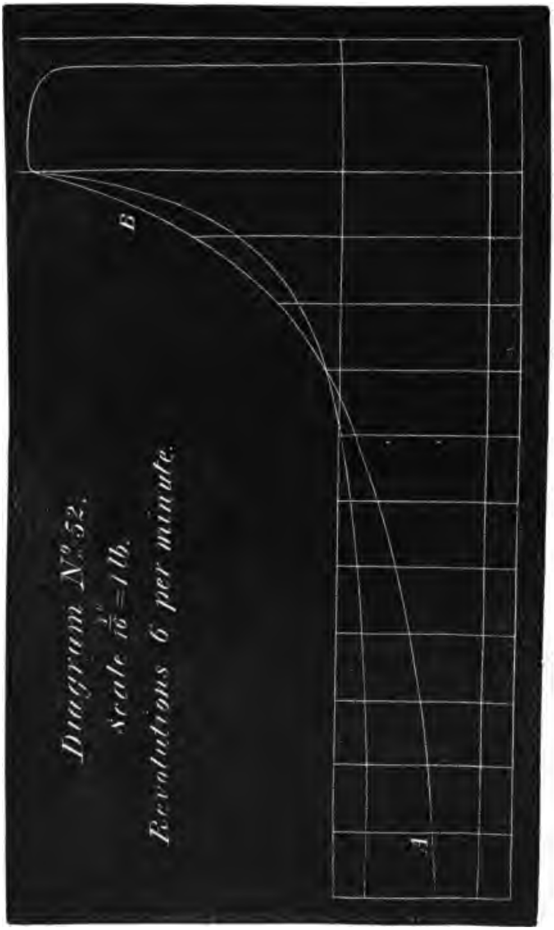
In considering this series of diagrams, one hardly knows whether to be most astonished at the amount of re-evaporation shown at one revolution per minute, or at the extent to which condensation continues after the cut-off, and re-evaporation still

appears, at 20 revolutions. Will a similar analysis of diagrams from unjacketed cylinders generally, of engines making this number of revolutions per minute, show the same indication?



It will not, for the following reasons: These cylinders were small, and the stroke was short. As the diameter of a cylinder is increased, the condensing surface presented by its walls increases directly as the diameter, but the area, and conse-

quently the volume of steam, increases as the square of the diameter. The condensing surface presented by the cylinder head and piston increases, however, as the square of the diameter, but again the volume of steam cut off at a given proportion of



the stroke increases directly as the length of the stroke. So the loss occasioned in this manner diminishes as the diameter and the stroke are increased. At the short stroke of these engines the mean piston speed at 20 revolutions is only 120 feet per

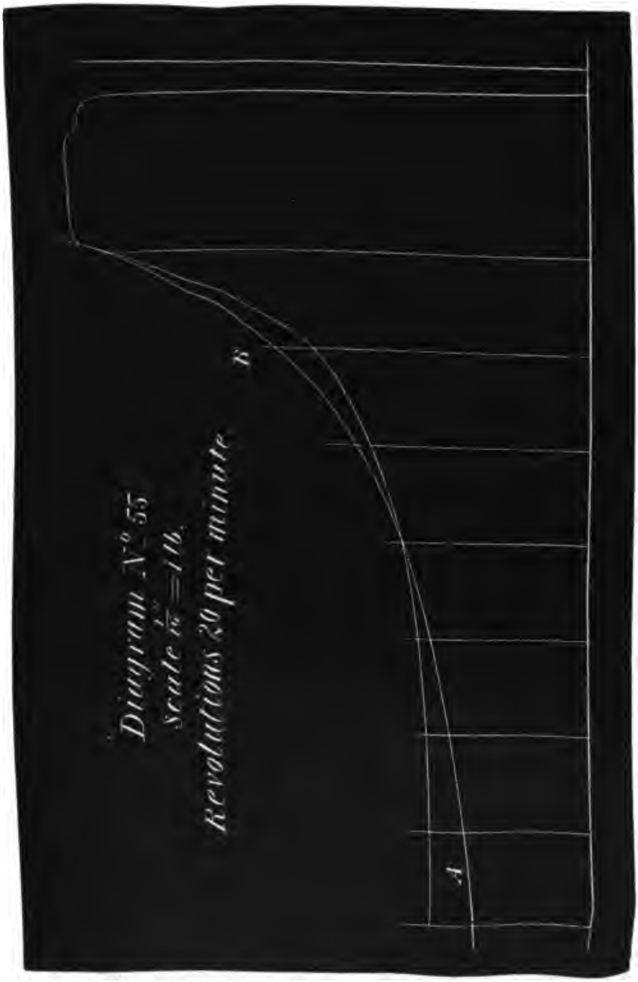
minute. Again, in unjacketed cylinders, in which the piston makes so few strokes per minute, the steam generally follows much farther than it does in these.

Abundant reason is seen, in the smaller diameters and shorter

stroke, why the loss in this case should be more aggravated than in the case of the *Michigan.*

It has been stated that the point of cut-off was fixed. This statement must be modified. The point of liberation was

fixed, and was identical in all the cases; but as its speed was increased, the piston moved farther while the port was being



closed, and so the point of cut-off became continually later, as shown.

These engines had, a few months previously, been tested by a commission appointed by the parties, and the Report of their

trials is published, but without any diagrams. From this Report it appears that the duty of these engines, when making ·866 of a revolution per minute, was 8,487,370 foot-pounds

with the consumption of 100 lbs. of coal (not 1 cwt.), and that when making 10·167 revolutions per minute it was 25,865,740 foot-pounds, with the consumption of 100 lbs. of coal.

At its usual speed, of about 50 revolutions per minute, with

5 feet to 7 feet stroke, giving a piston travel of from 500 to 700 feet per minute, this type of engine, unjacketed, has been regarded as quite economical; but at the above speeds the same engine, in perfect condition, and with the employment of all the mechanical means so much relied on to produce economy, gives at ·866 of a revolution per minute only one-twelfth of the duty attained in the best pumping-engines, which duty is increased threefold, or to one quarter of the best duty, by the mere increase of its speed to 10·167 revolutions per minute.

SECTION IV.

OF OTHER APPLICATIONS OF THE INDICATOR.

Although the principal use of the Indicator is to exhibit the behaviour of the steam in the cylinder of an engine, still this is by no means the only purpose to which it can be advantageously applied. Indeed so far is it from being limited to this use, that it affords the sole means of exhibiting and recording the changes of pressure which take place in any chamber in which an elastic or an inelastic fluid is confined. We will here briefly mention a few places where the Indicator ought to be regularly applied, and where it certainly will be applied by any one who shall ever add anything to his own knowledge, or that of others, on these subjects. These applications are mostly in connection with its use on the cylinder. Other uses of the instrument besides those here mentioned will present themselves in various branches of engineering.

On the Boiler.—The action of engines working steam expansively produces pulsations in the boiler, sometimes of a violent character. These can be exhibited only by the application of the Indicator. The difficulty of communicating the motion of the piston correctly to an instrument set on the boiler may, in most cases, be obviated by employing for this purpose a brass wire, as already recommended for general use. (*See* page 37.) These pulsations, the character and degree of which vary with the circumstances of each particular case, tend, sometimes very strongly, to increase the tendency of a boiler to prime, and sometimes, without doubt, they produce injurious or even destructive effects on the boiler itself. Very little is known about them; it seems certainly desirable that they should be better understood.

On the Steam-chest.—This application of the Indicator is of the first importance, in all cases in which the pressure realised in the cylinder is materially less than that existing in the boiler, or when it falls in the cylinder during the advance of

the piston, before the closing of the port. In such cases it is usual to jump to the conclusion that the port is too small, or the valve-travel is insufficient, when, for all that anybody *knows*, it may be that the fault lies wholly in the steam connections, and that no pressure materially greater than that shown in the cylinder existed at the same instants in the chest. Of course a flow of steam cannot, even under the most favourable conditions, be had without *some* difference of pressure; but the connection with the boiler ought to be sufficiently large to maintain in the chest, during all the while the port is open, a pressure approaching, certainly within one or two pounds, that existing in the boiler. The importance of an area of communication between the boiler and steam-chest, sufficient to prevent a fall of pressure in the latter while the steam is following the piston, is something that purchasers of engines sometimes fail properly to appreciate. We remember once advising a 6-inch steam-pipe. The party had a quantity of 4-inch pipe, which he had been intending to use. He listened, probably with some impatience, to our exposition of the subject, illustrated by imaginary diagrams traced on the wall, till, just as we thought we had brought conviction to his mind, he abruptly closed the interview with the exclamation: "Well, I would like to know where the economy of your engine is, if you want more steam than a 4-inch pipe can carry!"

Sometimes, on account of distance or unavoidable angles, it becomes difficult to maintain the pressure properly in the chest, without making the pipe of an objectionable size. In some cases of engines working at a high grade of expansion this difficulty has been met by making the pipe, for a short distance from the engine, large enough to serve also as a steam-reservoir, when the remainder may be made comparatively small; and through this the flow of steam, instead of being arrested by the closing of the port, goes on continuously, and at a rate approximately uniform. Of course nothing can be *known* as to the sufficiency or insufficiency of port opening, when a loss of pressure is shown on diagrams from the cylinder, unless one knows the pressure in the chest at each instant during the stroke.

The four following diagrams, drawn to the same scale, illustrate the action of the steam in the chest. The point of cut-off is marked on each by the letter *c*. In diagrams A and B a singular phenomenon is presented. When the current of steam

was arrested by the closing of the port, the pressure rose to a point above that in the boiler, and was maintained there in each case during about the same interval. The real boiler pressure is shown just previous to the opening of the ports.

No. 41.

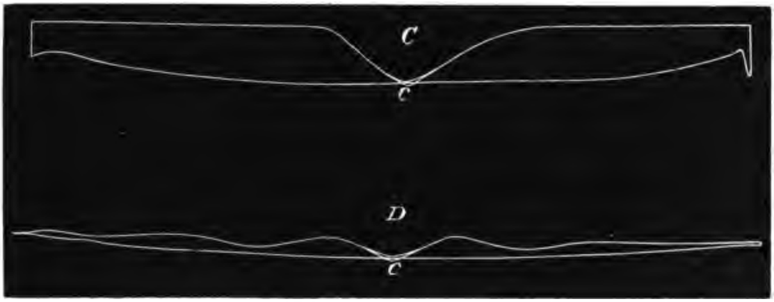
No. 42.

The entire action, as more fully shown in diagram A, is somewhat curious. The steam in the pipe refused to be put in motion instantly at a sufficient velocity to maintain the pressure, even though the motion of the piston was as yet scarcely sensible; it then flowed with such an augmenting velocity that

notwithstanding the increasing speed of the piston, the loss of pressure at the commencement of each stroke was nearly regained while the port remained open, when the momentum of the moving current had become so great that, after the closing of the port, a pressure considerably above that in the boiler had to be exerted in the chest, in order to arrest it. These two diagrams were taken from the "Allen" engine in the Paris Exposition, making 200 revolutions per minute, and connected by a short branch-pipe with the steam main. The rise of pressure in the chest above the boiler pressure, after the closing of the port, might be supposed to be a common occurrence, but it does not seem to be so. The writer has never observed it when the connection with the boiler was of uniform area.

Diagram C represents the loss of pressure caused by an in-

No. 43.



sufficiency of area in the steam-pipe. This diagram differs from the preceding ones in almost every respect. The loss of pressure was much greater, was much more nearly instantaneous at the commencement of the stroke, and increased as the piston advanced. After the closing of the port the pressure rose only to that in the boiler, at which point it remained steady during the remainder of the stroke.

Diagram D shows the behaviour of the steam in this chest, at the same speed of the piston, after an enlarged section of pipe had been introduced next to the engine, to serve as a reservoir of steam, as above recommended.

On the Exhaust-chamber. When a back pressure exists in the cylinder, the question to what it is due can only be determined

by applying the Indicator to the exhaust-chamber or passage. Then it will at once appear where the fault lies. If the same back pressure appears there, the fault is in the outlet; if, on the contrary, none should appear, then we can affirm certainly that the port opened for release is too small. Generally both contribute to the result. The Indicator should, for this test, be placed as near to the valve as possible. Elbows in exhaust-pipes will be found to be efficient causes of back pressure. The exact loss occasioned by a bend in the pipe can be shown by taking a diagram of pressure from each side of it.

On the Condenser.—In non-condensing engines the atmospheric line gives us the point from which to measure the avoidable back pressure on the piston, but in condensing engines we must obtain it by means of the Indicator placed on the condenser. The vacuum-gauge will, to be sure, enable us to measure the distance of this point from the atmosphere, if we are content to assume its correctness without proof; but the application of the Indicator will give us the line of pressure in the condenser, and that of the back pressure in the cylinder, on the same sheet, drawn by the same instrument, and the difference stands demonstrated. The quality of the gauge is also shown at the same time.

On the Air-Pump.—This application of the Indicator shows at once the nature of the performance of the air-pump, and the power required to operate it. There are air-pumps in use, the makers of which would probably alter their plans materially before constructing another one, if they knew what this silent servant stands ready to tell them.

On Pumps of every description.—The relation between the pressure in the pump-chamber and the velocity of the flow, as affected by a variety of causes, and that between this pressure and the power required to produce it, and the variations of pressure, and consequently of motion of the fluid, and the violence of the concussion, technically known as "the water-hammer"—all these the revelations of the Indicator show clearly. In fact, the instrument will often tell a great deal more than engineers want to know, or, at any rate, to have others know, about their work. In these applications of the Indicator the correction for temperature, as directed on page 48, must not be overlooked.

SECTION V.

OF THE USE OF THE TABLES OF THE PROPERTIES OF STEAM IN CALCULATING THE DUTY OF BOILERS.

To evaporate water of any given temperature into steam of any density, it is necessary to impart to it the number of thermal units contained in steam for that pressure, as given in Table IV., less the number which were at first contained in the water.

The duty of boilers is generally expressed by the number of pounds of water they will evaporate from a temperature of 212°, and under the atmospheric pressure, by the combustion of a pound of coal. In this case the temperature is not raised; the water merely passes from the liquid into the gaseous state, and the heat required to be imparted is that of vaporisation only. The number of thermal units necessary to produce this change of state diminishes very considerably as the temperature increases, being at 32°, 1091·7, at 212°, 965·7, and at a pressure of 210 lbs. on the square inch, and temperature of 385·671°, only 840·4.

Under the atmospheric pressure the total number of thermal units contained in the steam is ..	1178·6
Of which there are already contained in the water	212·9
Leaving to be imparted	965·7

The heat of combustion of one pound of carbon is 14·500 thermal units, which will evaporate from 212° 15 lbs. of water. ($14 \cdot 500 \div 965 \cdot 7 = 15$.)

In coals of good quality, *as an average*, the percentage of hydrogen, whose heat of combustion is 4·28 times that of carbon, will about compensate for the incombustible ingredients, so that a pound of good coal may be considered equivalent to a pound of carbon, and the above is its theoretical duty.* Four-

* For a full exposition of this subject see Rankine's 'Manual of the Steam Engine.' The American anthracites contain no hydrogen. Their steam-generating power seems precisely proportioned to their percentage of carbon.

fifths of this duty, or the evaporation of 12 lbs. of water by the combustion of a pound of coal, ought to be realised in practice.

We are to inquire how, in ordinary practice, evaporating from different temperatures, and under different pressures, the equivalent of this theoretical duty is to be ascertained. The following is the rule:—

The weight of water evaporated by the combustion of a pound of coal varies inversely as the quantity of heat necessary to be imparted.

Thus, to take an extreme case, let water at 32° be evaporated under a pressure, counting from perfect vacuum, of 120 lbs. on the square inch.

The number of thermal units contained in the steam is	1217·94
Number at first contained in the water	32·
Number necessary to be imparted ..	1185·94

1185·94 : 965·7 :: 12 : 9·77; which is, therefore, the equivalent number of pounds of water evaporated by the combustion of a pound of coal under these conditions.

In case of boilers whose duty is considerably less than the above, a gain can be effected by heating the feed-water up to the temperature existing in the boiler, by means of the waste heat in the gases; when the higher the pressure the greater, instead of the less, will be the number of pounds evaporated by the combustion of a pound of coal. This is because the latent heat only has afterwards to be supplied, the amount of which diminishes as the pressure is increased. The plan is, however, a mere extension of the heating surface of the boiler; but it has this advantage, that the heating surface of the apparatus is kept free from soot, while that of the boiler is not. In locomotive and other non-condensing engines, the temperature of the exhaust steam should be fully imparted to the feed-water, since, as we see from the Tables, in round numbers, each 9·5° so imparted saves one per cent. of the fuel.

The evaporative duty performed, in evaporating from feed-water of a given temperature into steam of any pressure, having been ascertained in the case of any boiler, the equivalent evaporation from 212°, and under the atmospheric pressure, is found by reversing the proportion illustrated above.

Example.—Let 9 lbs. of water be evaporated from a tempera-

ture of 130° into steam of 100 lbs. pressure, by the combustion of a pound of coal. Then:

The number of thermal units contained in the steam will be	1213·850
Of which there were at first contained in the water	130·192
Leaving to be imparted	1083·658

and 965·7 : 1083·658 : : 9 : 10·1; which is, therefore, the equivalent evaporation from 212° under the pressure of the atmosphere.

PART FIFTH.

SECTION I.

INTRODUCTORY.

We have thus far considered the action of steam in the cylinder of an engine, and the force exerted by it upon the piston, to produce and to resist its motion, and the distribution of these forces through the stroke, as the Indicator either reveals them to us or enables us to represent them. We have learned how, by means of this instrument, to measure these forces, to ascertain the quantity of steam consumed, and to demonstrate excellences and detect defects of principle or of construction; in fine, to investigate the action of the steam, and the operation and effect of every part of the distributing mechanism, from the boiler to the condenser.

Now another subject opens before us, one which is intimately connected with, and primarily dependent upon, the revelations of the Indicator, and the understanding of which will give completeness to our knowledge of the steam-engine as a prime mover, of which the Indicator itself imparts to us only the beginning.

Usually in steam-engines the piston is caused to move forward and backward by the alternate preponderance of pressure on its opposite sides, for the purpose of producing rotary motion, through the medium of the connecting-rod and the crank; and we purpose now to trace through these connections the force exerted by the steam in the cylinder, until we shall see it given off from the shaft.

The subject naturally divides into two parts.

We shall inquire, first, into the pressure exerted on the crank; and, second, into the rotative effect which this pressure produces.

The former we shall find to be the result of the combined action of the force of the steam and the inertia of those parts of

the engine through which this is transmitted. The latter we shall find to vary according to the angle with the line of centres at which the crank receives the pressure.

Although, since we shall disregard the loss of power by friction the *areas* of all the diagrams of pressure, whether on the piston, or on the crank, or on whatever opposes the rotation of the shaft, will be identical, it will be interesting to observe how their *forms* will differ, and how much farther away we shall get, at each step, from the figure described by the Indicator.

This inquiry will not only be interesting in itself, extending considerably our previous knowledge of the subject, but it will also prove important in its immediate practical results, enabling us to solve the problem how, in a single-cylinder engine, and without a large or heavy fly-wheel, to work steam of high pressure, and at a high grade of expansion, and still to maintain the closest approach to uniform rotation.

SECTION II.

OF THE PRESSURE ON THE CRANK WHEN THE CONNECTING-ROD IS CONCEIVED TO BE OF INFINITE LENGTH.

The diagram represents the pressure of steam on the piston at each point of its stroke. It is commonly supposed that this is also a representation of the corresponding pressure on the crank. Such, however is not by any means the case. The parts of an engine whose office it is to transmit motion to the crank—namely, the piston and rod, cross-head, beam and connecting-rod, called collectively the transmitting parts, or the reciprocating parts, or parts having alternate movements—receive first the force of the steam. Their own inertia must be overcome before any power can be communicated through them. This varies according to their weight, and to the square of the velocity imparted to them in moving through a given distance; so that to put them in motion may in one case require but little power, and in another it may, as we shall see, absorb the entire energy of the steam; nay, this may even be insufficient, and often it is insufficient, for this purpose, and the living force of the fly-wheel is required, in aid of the steam-pressure, at the commencement of the stroke, to impart to them their motion.

The acceleration of the motion of the reciprocating parts takes place during the first half of the stroke; at the mid-stroke they have attained their full velocity; during the latter half of the stroke they must impart their living force to the crank, the resistance of which brings them to rest.

To ascertain, therefore, the pressure exerted on the crank at each point in its semi-revolution, we must subtract from the Indicator diagram during the first half of the stroke, and add to it during the latter half, the force required to be exerted at each point, by the steam and the crank alternately, to overcome the inetria of these parts,—inertia being the disposition of matter to continue in the state, whether of rest or motion, in which it is. It is thus obvious that, especially in the case of swift-running engines, unless this action be understood, we can have no correct

idea of the manner in which the force of the steam is applied to the crank.

In examining this subject the horizontal form of engine will be employed for illustration, as being better suited to the purpose than the beam-engine, because in it all the reciprocating parts have the same motion. It is, however, to be observed, that the action we are to investigate is in all balanced vertical engines identical with that in horizontal engines, except only as it is diminished by the lesser velocity of the centres of oscillation of the beam. In unbalanced vertical engines, on the down-stroke the action of gravity reduces the force of the steam required to give to these parts their motion, and increases the resistance of the crank required to arrest it, and on the up-stroke produces the contrary effect. With this modification, the action under discussion is the same also in engines of that class.

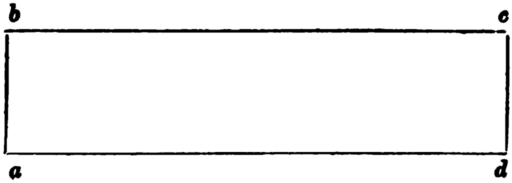
We shall, in this section, disregard the effect of the angular vibration of the connecting-rod, and shall consider the motion of the piston as if it were the same at corresponding points in the opposite strokes.

The reciprocating parts of a horizontal engine are four; namely, the piston, piston-rod, cross-head, and connecting-rod. These, taken together, are properly regarded as a projectile. We shall consider them as such, and also as moving without friction. At the commencement of a stroke our projectile is at rest. At the middle of the stroke it has attained its greatest velocity, which is equal to that of the extremity of the crank. This is much greater than its mean velocity, being to the latter as $\frac{\pi}{2}$ is to 1, or as 1·5708 : 1. At the termination of the stroke it is at rest again.

Confining our attention, for the present, to the action which takes place during the first half of the stroke, we inquire: How is this velocity, which this mass attains at the middle of the stroke, imparted to it? Two facts are before us: first, it does acquire this velocity; and second, it acquires it in moving through one-half of its stroke, from a state of rest. A force is required, acting within this distance, to impart this velocity. What is that force?

We may conceive the motion of the mass to be uniformly accelerated by a constant force acting through this distance. Then if, in the following figure, the vertical line *a b* represents the force and the horizontal line *a d* represents the half-stroke;

the rectangle *a b c d* will represent the work done by such constant force.



We know very well that this cannot be a correct representation of the accelerating force as it is actually exerted at each point in the half-stroke, because no such abrupt cessation of it takes place at the middle of the stroke; but if this figure represents the work done by a constant force, which by acting through this distance *would* impart this final velocity, then the actual forces, however they may vary in amount at different points in this distance, must be the equivalent of such constant force: the same work is done, and it must be represented by some figure of equal area, whatever may be its form.

We will therefore inquire, first, What *constant* force, acting through this distance, would impart to this mass its final velocity? This force is found by comparing the velocity acquired with that which a falling body acquires in falling through the same distance from a point of rest.

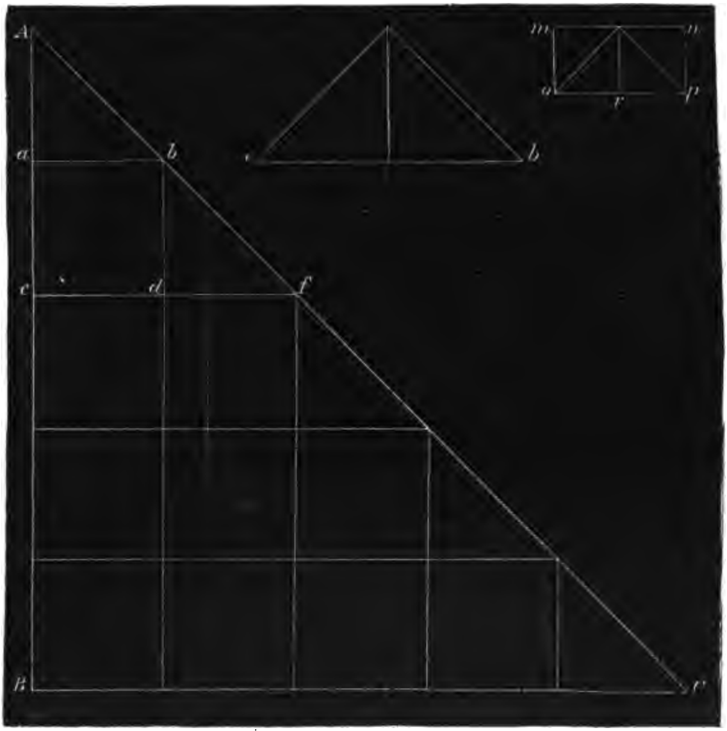
Before doing this we will endeavour, for the benefit of readers not already familiar with the subject, to state, as briefly as possible, the laws of uniformly accelerated motion, as illustrated in falling bodies.

A falling body always in reality meets the resistance of the atmosphere, but theoretically it is supposed to be moving without resistance. The force which urges it is the attraction of the earth. This force imparts to all bodies the same motion, since it varies directly as the mass. So the attracting force is always equal to the weight of the body. It has been established that this force near the level of the sea, in this latitude, will cause a body to fall, *in vacuo*, from a point of rest, 16·083 feet in one second, at the end of which time it has acquired a velocity of 32·166 feet per second.

The motion of a falling body is illustrated in the following figure, No. 44:—Let the height A B of the right-angled triangle A B C represent five seconds of time, and let the

inclination of the hypothenuse A C represent the uniform acceleration of the motion of a falling body, which is produced by the constant action of the force of gravity. Then the base B C, and the four lines parallel with it, and dividing the height into five equal parts, will represent the velocity acquired by the body at the end of each second, and the areas of these five

No. 44.



divisions of the triangle will represent the distances through which the body falls in the several seconds. Let a body fall from the point A. The area of the triangle A *a b* represents the distance, 16·083 feet, through which it will fall during the first second, and the base *a b* represents the velocity 32·166 feet per second, which it will acquire in falling through that distance. If now the attraction of the earth could cease, the body would continue to fall with the velocity already

imparted to it, and would, during the next and each succeeding second, fall 32·166 feet, or twice as far as it fell during the first second, as is illustrated by the square *a b c d*, and those below it.

The attraction of the earth, however, cannot cease, but continues to urge the body precisely as at first, so that the distance fallen through during the second second is, not twice, but three times that fallen through during the first, and so on, as shown by the figure.

If we represent the line *a b* by 2, then the succeeding base-lines will be represented by the even numbers, 4, 6, 8, 10, showing the velocity, in feet per second, acquired at the expiration of the successive seconds to be as follows:—

At the end of one second	..	32·166	feet per second.	
„ „ two seconds	..	64·332	„	„
„ „ three „	..	96·498	„	„
„ „ four „	..	128·664	„	„
„ „ five „	..	160·830	„	„

If, in like manner, we represent the area A *a b* by 1, then the successive areas below it will be represented by the odd numbers, 3, 5, 7, 9, showing the distances fallen through in the successive seconds to be as follows:—

During the first second	.. ..	16·083	feet.
„ second „	.. ..	48·249	„
„ third „	.. ..	80·415	„
„ fourth „	.. ..	112·581	„
„ fifth „	.. ..	144·747	„

It will be seen, also, that the aggregate areas, representing the total distances fallen through, from the point of rest to the end of each second, are represented by the numbers 1, 4, 9, 16, 25, which are the squares of the seconds, showing the total distances fallen through to be as follows:—

In one second	.. .. ..	16·083	feet.
„ two seconds	.. .. ..	64·332	„
„ three „	.. .. ..	144·747	„
„ four „	.. .. ..	257·328	„
„ five „	.. .. ..	402·075	„

In the following Table all these relations are presented in one view, the time being extended to ten seconds. It will be observed that, for each second, the mean velocity is 16·083 feet per second less than the terminal velocity.

TABLE XI.

Exhibiting the Acceleration of the Motion of a falling Body during ten seconds.

Seconds occupied in falling.	(a) Total distance fallen through, in feet.	(b) Differences; being the distance fallen through, or the mean velocity, during each second.	(c) 2nd Differences; being the constant acceleration, in feet per second.	(d) Velocity, in feet per second, acquired at the expiration of each second.
1	16·083	16·083	32·166	32·166
2	64·332	48·249	32·166	64·332
3	144·747	80·415	32·166	96·498
4	257·328	112·581	32·166	128·664
5	402·075	144·747	32·166	160·830
6	578·988	176·913	32·166	192·996
7	788·067	209·079	32·166	225·162
8	1029·312	241·245	32·166	257·328
9	1302·723	273·411	82·166	289·494
10	1608·300	305·577	32·166	321·660

From this Table the following law is deduced:—The velocities acquired by falling bodies are directly as the times of falling, and the distances fallen through to acquire those velocities are as the squares of the times.

$v = at$

$h = \frac{at^2}{2}$

In this inquiry, however, we have to banish the idea of time from our minds, and to consider only the relation of velocities acquired and distances fallen through in acquiring them.

We observe that at the end of one second the velocity per second acquired is twice the distance fallen through. Now there is nothing peculiar about a second, but it is universally true of a falling body that, in any interval of time whatever, it acquires a velocity, *per such interval*, which is double the distance through which it has fallen, or, that its final velocity is twice its mean velocity; and this is the simple law of uniformly accelerated motion.

But, for purposes of comparison, some interval must be agreed on as a unit of time, and so the velocity of falling bodies is expressed in feet per second. It follows that velocities per units smaller than one second are enlarged, and those per larger units are diminished, in expression. For example: at the end of half a second we do not say that a falling body has acquired a velocity

of 8·0415 feet per half-second, which is twice the distance it has fallen, but that it has acquired a velocity of 16·083 feet per second; and at the end of two seconds we do not say that it has acquired a velocity of 128·664 feet per two seconds, which also is twice the distance it has fallen, but that it has acquired a velocity of 64·332 feet per second. Time, one of the two factors whose changes kept the velocity acquired in apparently a constant ratio with the distance fallen through, being now made unchangeable, the other factor must vary as the square roots of the distances, and so we have the law:—

The velocities per second acquired by a falling body vary as the square roots of the distances fallen.

Whence

$$v = 8 \cdot 0207 \sqrt{d}$$

Now if a body is being impelled in a horizontal direction its own weight is neutral—has no tendency to produce motion, nor, except through friction, which is excluded by our terms, to arrest it. Therefore a constant force equal to its weight must be required to impart to it the same velocity that it would acquire in falling. We will hereafter, then, consider our projectile as substituted in the place of a falling body.

There is, however, this important difference. In the case of falling bodies the force is invariable, and either the distance fallen through or the velocity acquired is given to find the other. But in the action we have to consider it is the force which is the unknown quantity. The distance moved through, and the velocity acquired, are both known, and the force is sought, greater or less than that of gravity, or the weight of the mass, which is required to impart the velocity by acting through the distance.

The determination of this force is a very simple matter. It has been established, both by mathematical investigation and by direct experiment, that forces act cumulatively; that is, that, for example, one-half the force of gravity will impel a body one-half as far, twice that force twice as far, four times that force four times as far, in a given interval of time; and the velocity acquired per such interval is twice the distance moved through, or, the velocities per second vary as the square roots of the distances, precisely as when the force is that of gravity.

For the purpose of imparting velocity, then, force is the equivalent of distance. One may be substituted in place of the other in any degree, or, to impart a given velocity, it is necessary

only that the product of the force into the distance shall be a constant quantity,—either factor may be diminished as much as the other is increased. Thus, for example, force 1 must act through distance 4 to impart a twofold velocity, and force 2 acting through distance 2, or force 4 acting through distance 1, will impart the same velocity.

The distances through which a given force must act, to impart different velocities, vary as the squares of the velocities imparted; and so the forces, acting through a given distance, to impart different velocities, must vary in the same manner.

We have now considered the constant acceleration of motion. A few words will explain its retardation. The inertia of a moving body, or the force with which it opposes being brought to rest, is the same as that with which a body at rest resists being put in motion; but it is commonly distinguished from the latter by the term 'momentum.' The momentum of a moving body, or its *vis viva*, or living force, or the work stored up in it, is the product of the force which it exerts, multiplied by the distance through which it will move while exerting it, and, less the loss of power by friction, it is equal to the force which put it in motion multiplied by the distance through which that was exerted. So that, if bodies of the same weight are moving with different velocities, the resistance which will bring each to rest, or the distance through which that resistance is exerted, or the product of the two, must differ as the squares of the velocities.

We arrive, then, at this general law:—

Uniformly accelerating or retarding force varies, directly as the mass, inversely as the distance within which a given velocity is imparted or arrested, and as the square of the velocity imparted or arrested within a given distance.

In applying this law to the case of steam-engines:—

Let W = the weight of the transmitting parts;
g = the velocity, in feet per second, acquired by a falling body in one second = 32·166;
D = the distance, 16·083 feet, through which a body falls in one second, or in acquiring the velocity g;
v = the maximum velocity attained by the transmitting parts at the middle of the stroke;
d = the distance—one half the stroke—through which these parts move in acquiring this velocity; and,
f = the constant force required.

Work = $fd = \frac{Wv^2}{2g}$ $\quad f = \frac{Wv^2}{2gd}$

Then $f = W \times \frac{D}{d} \times \frac{v^2}{g^2}$; whence, by cancelling the constant quantities, D and g^2, we obtain the formula $f = W \times \frac{v^2}{64 \cdot 332d}$

What, then, would be the constant force required to impart their motion to the transmitting parts of an engine of 24 ins. diameter, or 452 sq. ins. area, of cylinder, by 5-feet stroke, and making 60 revolutions per minute, and the transmitting parts of which weigh 2000 lbs.?

The mean piston-speed is 600 feet per minute; the velocity attained by the transmitting parts at the mid-stroke is $600 \times 1 \cdot 5708 = 942 \cdot 5$ feet per minute, or $15 \cdot 708$ feet per second, and we have,

$$f = 2000 \times \frac{246 \cdot 74}{160 \cdot 83} = 3068 \text{ lbs.}$$

or $6 \cdot 8$ lbs. on each 1 sq. in. of piston area.

Again, required the constant force, which would impart their velocity to the transmitting parts of an engine of 16 ins. diameter, or 201 sq. ins. area, of cylinder, by 30 ins. stroke, and making 140 revolutions per minute, and in which the weight of these parts is 1100 lbs.

The mean piston-speed is 700 feet per minute, the velocity attained by the transmitting parts at the mid-stroke is 1100 feet per minute, or $18 \cdot 333$ feet per second, and we have,

$$f = 1100 \times \frac{336 \cdot}{80 \cdot 415} = 4596 \text{ lbs.}$$

or 23 lbs. on each 1 sq. in. of piston area.

These, then, are the forces in the above two cases respectively, which, supposing them to be constantly exerted, would be required during the first half of each stroke in one direction, to give to the transmitting parts of the engine their velocity; and again, during the latter half of each stroke in the opposite direction, to bring them to rest.

We do not care to hasten on in our inquiry. The law of uniformly accelerated or retarded motion is one so interesting in itself, the idea of any action of this nature taking place in a steam-engine is so novel to almost every one, and it is so difficult at once to realise the fact of such considerable forces being required for this purpose, while, at the same time, a clear comprehension of it is so necessary to a proper understanding of that which lies before us, that we would dwell upon it for a

moment, and observe it from different points of view, and become, as it were, familiar with it.

Let us, then, return to the example last cited. The velocity attained at each mid-stroke is 18·333 feet per second. How far would a body need to fall to acquire this velocity? The answer is at hand. The velocities vary as the square roots of the heights, and $32·166 : 18·333 :: \sqrt{16·083} : \sqrt{5·2235}$. 5·2235 feet is, therefore, the distance through which a body impelled by a constant force equal to its own weight must move to acquire this velocity. But the distance in which the transmitting parts do in fact acquire this velocity is only 1·25 feet. How much greater, then, than their own weight must the accelerating force be? The forces are inversely as the distances, and we have, $\frac{1100 \times 5·2235}{1·25} = 4596$ lbs. as before.

Again, what velocity would a falling body acquire in falling through 1·25 feet from a state of rest?

Answer: $32·166 \frac{\sqrt{1·25}}{\sqrt{16·083}} = 8·968$ feet per second.

But the velocity actually acquired by these parts in moving through this distance is 18·333 feet per second. The force required varies as the square of the velocity, and we have

$$\frac{1100 \times 18·333^2}{8·968^2} = 4596 \text{ lbs., as before.}$$

So the corrrectness of our formula is proved, and we see clearly that to give to these parts the velocity which they attain at each mid-stroke, and then to bring them to rest, this great amount of force would need to be constantly exerted.

But it is quite certain that no such action as this uniform acceleration and retardation, with an abrupt transition from one to the other at the mid-stroke, takes place in a steam-engine. The amount of work which they represent is, however, done. The acceleration and retardation do take place. Evidently it must be in some ratio very different from a constant one. What is that ratio?

The following Tables have been prepared for the purpose of presenting a complete answer to this question:—

TABLE XII.

EXHIBITING THE ACCELERATION OF THE PISTON OF A STEAM-ENGINE AT THE COMMENCEMENT AND AT THE TERMINATION OF EACH INTERVAL OF 1° OF THE QUARTER-REVOLUTION OF THE CRANK, COUNTING FROM THE LINE OF CENTRES.

Angular Motion of the Crank.	(a) Versed sine of the angle; or total distance through which the piston moves, from the commencement of the stroke to the end of each degree.	* (b) Differences of the versed sines; being the motion of the piston, or its mean velocity, during each degree.	† (c) Differences of the mean velocities: being the acceleration per degree at the commencement of the stroke and at the termination of each degree.	(d) Differences of the accelerations; being the diminution of the acceleration during each degree.	(e) Differences of the diminutions: being the increase in the diminution of acceleration during each degree.
0°	·0000000000		·0003046096		
1	·0001523048	·0001523048	·0003045633	·0000000463	·0000000928
2	·0006091729	·0004568681	·0003044242	·0000001391	·0000000929
3	·0013704652	·0007612923	·0003041922	·0000002320	·0000000925
4	·0024359497	·0010654845	·0003038677	·0000003245	·0000000927
5	·0038053019	·0013693522	·0003034505	·0000004172	·0000000922
6	·0054781046	·0016728027	·0003029411	·0000005094	·0000000926
7	·0074538484	·0019757438	·0003023391	·0000006020	·0000000919
8	·0097319313	·0022780829	·0003016452	·0000006939	·0000000918
9	·0123116594	·0025797281	·0003008595	·0000007857	·0000000919
10	·0151922470	·0028805876	·0002999819	·0000008776	·0000000910
11	·0183728165	·0031805695	·0002990133	·0000009686	·0000000916
12	·0218523993	·0034795828	·0002979531	·0000010602	·0000000903
13	·0256299352	·0037775359	·0002968026	·0000011505	·0000000906
14	·0297042737	·0040743385	·0002955615	·0000012411	·0000000900
15	·0340741737	·0043699000	·0002942304	·0000013311	·0000000898

16	·0387383041	·0046641304	·0002928095	·0000014209	·0000000888
17	·0436952440	·0049569399	·0002912998	·0000015097	·0000000891
18	·0489434837	·0052482397	·0002897010	·0000015988	·0000000881
19	·0544814244	·0055379407	·0002880141	·0000016869	·0000000877
20	·0603073792	·0058259548	·0002862395	·0000017746	
		·0061121943			·0000000872
21	·0664195735	·0063965720	·0002843777	·0000018618	·0000000869
22	·0728161455	·0066790010	·0002824290	·0000019487	·0000000855
23	·0794951465	·0069593958	·0002803948	·0000020342	·0000000857
24	·0864545423	·0072376707	·0002782749	·0000021199	·0000000850
25	·0936922130	·0075137407	·0002760700	·0000022049	·0000000837
26	·1012059537	·0077875221	·0002737814	·0000022886	·0000000836
27	·1089934758	·0080589313	·0002714092	·0000023722	·0000000825
28	·1170524071	·0083278858	·0002689545	·0000024547	·0000000823
29	·1253802929	·0085943033	·0002664175	·0000025370	·0000000807
30	·1339745962		·0002637998	·0000026177	
		·0088581031			·0000000807
31	·1428326993	·0091192045	·0002611014	·0000026984	·0000000793
32	·1519519038	·0093775282	·0002583237	·0000027777	·0000000788
33	·1613294320	·0096329954	·0002554672	·0000028565	·0000000778
34	·1709624274	·0098855283	·0002525329	·0000029343	·0000000770
35	·1808479557	·0101350499	·0002495216	·0000030113	·0000000758
36	·1909830056	·0103814844	·0002464345	·0000030871	·0000000754
37	·2013644900	·0106247564	·0002432720	·0000031625	·0000000738
38	·2119892464	·0108647921	·0002400357	·0000032363	·0000000731
39	·2228540385	·0111015184	·0002367263	·0000033094	·0000000724
40	·2339555569		·0002333445	·0000033818	
		·0113348629			·0000000709
41	·2452904198	·0115647547	·0002298918	·0000034527	·0000000699
42°	·2568551745		·0002263692	·0000035226	·0000000692

TABLE XII.—*continued.*

Angular Motion of the Crank.	(a) Versed sine of the angle; or total distance through which the piston moves, from the commencement of the stroke to the end of each degree.	(b) Differences of the versed sines: being in the motion of the piston, or its mean velocity, during each degree.	(c) Differences of the mean velocities: being the acceleration per degree at the commencement of the stroke and at the termination of each degree.	(d) Differences of the accelerations: being the diminution of the acceleration during each degree.	(e) Differences of the diminutions: being the increase in the diminution of acceleration during each degree.
43°	·2686462984	·0117911239	·0002227774	·0000035918	·0000000678
44	·2806601997	·0120139013	·0002191178	·0000036596	·0000000666
45	·2928932188	·0122330191	·0002153916	·0000037262	·0000000656
46	·3053416295	·0124484107	·0002115998	·0000037918	·0000000649
47	·3180016400	·0126600105	·0002077431	·0000038567	·0000000626
48	·3308693936	·0128677536	·0002038238	·0000039193	·0000000626
49	·3439409710	·0130715774	·0001998419	·0000039819	·0000000606
50	·3572123903	·0132714193	·0001957994	·0000040425	
51	·3706796090	·0134672187	·0001916970	·0000041024	·0000000599
52	·3843385247	·0136589157	·0001875364	·0000041606	·0000000582
53	·3981849768	·0138464521	·0001833188	·0000042176	·0000000570
54	·4122147477	·0140297709	·0001790450	·0000042738	·0000000562
55	·4264235636	·0142088159	·0001747170	·0000043280	·0000000542
56	·4408070965	·0143835329	·0001703356	·0000043814	·0000000534
57	·4553609650	·0145538685	·0001659023	·0000044333	·0000000519
58	·4700807358	·0147197708	·0001614185	·0000044838	·0000000505
59	·4849619251	·0148811893	·0001568856	·0000045329	·0000000491
60	·5000000000	·0150380749	·0001523049	·0000045807	·0000000478
61	·5151903798	·0151903798	·0001476776	·0000046273	·0000000466
62	·5305284372	·0153380574	·0001430056	·0000046720	·0000000447
					·0000000436

63	·5460095002	·0154810630	·0001382900	·0000047156	·0000000423
64	·5616288532	·0156193530	·0001335321	·0000047579	·0000000407
65	·5773817383	·0157528851	·0001287335	·0000047986	·0000000389
66	·5932633569	·0158816186	·0001238960	·0000048375	·0000000380
67	·6092688715	·0160055146	·0001190205	·0000048755	·0000000362
68	·6253934066	·0161245351	·0001141088	·0000049117	·0000000348
69	·6416320505	·0162386439	·0001091623	·0000049465	·0000000331
70	·6579798567	·0163478062	·0001041827	·0000049796	·0000000320
71	·6744318456	·0164519889	·0000991711	·0000050116	·0000000298
72	·6909830056	·0165511600	·0000941297	·0000050414	·0000000291
73	·7076282953	·0166452897	·0000890592	·0000050705	·0000000269
74	·7243626442	·0167343489	·0000839618	·0000050974	·0000000256
75	·7411809549	·0168183107	·0000788388	·0000051230	·0000000240
76	·7580781044	·0168971495	·0000736918	·0000051470	·0000000226
77	·7750489457	·0169708413	·0000685222	·0000051696	·0000000213
78	·7920883092	·0170393635	·0000633319	·0000051903	·0000000193
79	·8091910046	·0171026954	·0000581223	·0000052096	·0000000177
80	·8263518223	·0171608177	·0000528950	·0000052273	·0000000163
81	·8435655350	·0172137127	·0000476514	·0000052436	·0000000144
82	·8608268991	·0172613641	·0000423934	·0000052580	·0000000128
83	·8781306566	·0173037575	·0000371226	·0000052708	·0000000113
84	·8954715367	·0173408801	·0000318405	·0000052821	·0000000100
85	·9128442573	·0173727206	·0000265484	·0000052921	·0000000078
86	·9302435263	·0173992690	·0000212485	·0000052999	·0000000066
87	·9476640438	·0174205175	·0000159420	·0000053065	·0000000047
88	·9651005033	·0174364595	·0000106308	·0000053112	·0000000035
89	·9825475936	·0174470903	·0000053161	·0000053147	·0000000014
90°	1·0000000000	·0174524064	·0000000000	·0000053161	

TABLE XIII.

EXHIBITING THE ACCELERATION OF THE PISTON, AT THE COMMENCEMENT AND AT THE TERMINATION OF EACH INTERVAL OF 6′ IN THE FIRST THREE DEGREES, AND THE NINETIETH DEGREE OF THE QUARTER REVOLUTION OF THE CRANK, COUNTING FROM THE LINE OF CENTRES.

Angular motion of the crank.	(a) Versed sine of the angle; or total distance through which the piston moves, from the commencement of the stroke to the end of each tenth of a degree.	(b) Differences of the versed sines; being the motion of the piston, or its mean velocity, during each tenth of a degree.	(c) Differences of the mean velocities; being the acceleration, per tenth of a degree, at the commencement of the stroke, and at the termination of each tenth of a degree.	(d) Differences of the accelerations; being the diminution of the acceleration during each tenth of a degree.	(e) Differences of the diminutions; being the increase in the diminution of acceleration during each tenth of a degree.
0° 0′	·000000000000000	·000001523086712	·000003046173424	·000000000004638	·000000000009283
6	·000001523086712	·000004569255498	·000003046168786	·000000000013921	·000000000009275
12	·000006092342210	·000007615410363	·000003046154865	·000000000023196	·000000000009281
18	·000013707752573	·000010661542032	·000003046131669	·000000000032477	·000000000009280
24	·000024369294605	·000013707641224	·000003046099192	·000000000041757	·000000000009277
30	·000038076935829	·000016753698659	·000003046057435	·000000000051034	·000000000009279
36	·000054830634488	·000019799705060	·000003046006401	·000000000060313	·000000000009279
42	·000074630339548	·000022845651148	·000003045946088	000000000000069592	·000000000009279
48	·000097475990696	·000025891527644	·000003045876496	·000000000078871	·000000000009276
54	·000123367518340	·000028937325269	·000003045797625	·000000000088147	
1° 00′	·000152304843609		·000003045709478		
1° 6′	·000184287878356	·000031983034747	·000003045612052	·000000000097426	·000000000009279
12	·000219316525155	·000035028646799	·000003045505348	·000000000106704	·000000000009278
18	·000257390677302	·000038074152147	·000003045389368	·000000000115980	·000000000009267
24	·000298510218817	·000041119541515	·000003045264111	·000000000125257	·000000000009277
30	·000342675024443	·000044164805626	·000003045129577	·000000000134534	·000000000009277
36	·000389884959646	·000047209935203	·000003044985768	·000000000143809	·000000000009275

42	·000440139880617	·000050254920971	·000003044832861	·000000000153087	·000000000009278
48	·000493439634269	·000053299753652	·000003044670322	·000000000162359	·000000000009272
54	·000549784058243	·000056344423974	·000003044498688	·000000000171634	·000000000009275
2° 00′	·000609172980905	·000059388922662	·000003044317777	·000000000180911	·000000000009277
2° 6′	·000671606221344	·000062433240439	·000003044127596	·000000000190181	·000000000009270
12	·000737083589379	·000065477368035	·000003043928141	·000000000199455	·000000000009274
18	·000805604885555	·000068521296176	·000003043719412	·000000000208729	·000000000009274
24	·000877169901143	·000071565015588	·000003043501412	·000000000218000	·000000000009271
30	·000951778418143	·000074608517000	·000003043274143	·000000000227269	·000000000009269
36	·001029430209286	·000077651791143	·000003043037602	·000000000236541	·000000000009272
42	·001110125038031	·000080694828745	·000003042791791	·000000000245811	·000000000009270
48	·001193862658567	·000083737620536	·000003042536711	·000000000255080	·000000000009269
54	·001280642815814	·000086780157247	·000003042272366	·000000000264345	·000000000009265
3° 00′	·001370465245427	·000089822429613	·000003041998749	·000000000273617	·000000000009272
89° 00′	·982547593562717	·001745035962407	·000000053163055	·000000005315685	·000000000000158
6	·984292682688179	·001745089125262	·000000047847212	·000000005315843	·000000000000143
12	·986037819660853	·001745136972674	·000000042531226	·000000005315986	·000000000000139
18	·987782999164753	·001745179503900	·000000037215101	·000000005316125	·000000000000105
24	·989528215883754	·001745216719001	·000000031898871	·000000005316230	·000000000000100
30	·991273464501626	·001745248617872	·000000026582541	·000000005316330	·000000000000082
36	·993018739702039	·001745275200413	·000000021266129	·000000005316412	·000000000000064
42	·994764036168581	·001745296466542	·000000015949653	·000000005316476	·000000000000046
48	·996509348584776	·001745312416195	·000000010633131	·000000005316522	·000000000000037
54	·998254671634102	·001745323049326	·000000005316572	·000000005316559	·000000000000013
90° 00	1·000000000000000	·001745328365898	·000000000000000	·000000005316572	

These Tables exhibit, in decimals of the length of the crank,

First. The total motion of the piston* from the commencement of the stroke;

Second. The motion, or mean velocity, during each equal interval of time taken; and,

Third. The acceleration at the termination of each of these intervals.

These are given in the first Table for intervals of 1° through the quarter revolution of the crank; and in the second Table for intervals of one-tenth of 1° through the first three degrees, and the last degree of the quarter revolution.

The motion of the crank is supposed to commence at zero on the line of centres, and to be uniform; and so the degrees are taken to mark equal divisions of time.

First Column.

During the first half of its stroke the motion of a piston controlled by a crank is, disregarding, as we now do, the effect produced by the angular vibration of the connecting-rod, equal to the versed sine of the angle which the crank forms with the line of centres. In these Tables, the first column (*a*) gives the versed sines of the intervals taken, or the total distances through which the piston has moved from the commencement of the stroke, and corresponds with column (*a*) in Table No. XI. on page 203 of the motion of a falling body. It must be observed that this column tells nothing of the velocity of the piston, the versed sines merely represent the total distances through which it has moved from the point of rest.

Second Column.

The velocity of the piston varies as the sine of the angle.

The second column (*b*) gives the differences of the versed sines. These, it is evident, express the motion, or mean velocities, of the piston while the crank is traversing the intervals of arc taken, corresponding with the differences in column (*b*) of Table No. XI. The motion of the piston per degree, it will be seen, increases from ·0001523048 during the

* For convenience of expression, "the piston" will hereafter be used to represent the entire mass of the transmitting parts of the engine.

first degree of the motion of the crank from the line of centres to ·0174524064 during the 90th degree. The latter approaches very nearly to the constant velocity, per degree, of the crank in its arc, which, if radius = 1, is

$$\frac{2\pi}{360} \text{ or } \frac{2 \times 3{\cdot}14159265359}{360} = {\cdot}0174532925.$$

The mean velocity, per degree, of the piston for the last one-tenth of the 90th is ·0174532836, which agrees with that of the crank up to the eighth place of decimals. The Table gives this velocity per one-tenth of one degree, which is one-tenth of the above.

The differences of the versed sines vary directly as the sines. The curve representing the acceleration of the piston's motion approximating nearly to a straight line, we may, without sensible error, take the mean velocity of the piston during any short interval to be its velocity at the middle point of such interval. Now, if we divide the sine of any degree, $- 30'$, by the mean velocity during that degree, as found in column (b) of Table No. XII. we obtain always the quotient 57·2965, which expresses the constant ratio between the difference of the versed sines of any two consecutive degrees and the sine of the intermediate angle.

Therefore the velocity of the piston varies as the sine of the angle which the crank forms with the line of centres. This relation is of such consequence, that a knowledge of it is the key to the understanding of the crank motion, as we shall see hereafter.

These Tables give the mean velocity during each 1·°, or ·1°. In order, therefore, to get the velocity at the end of any such interval, one-half the acceleration for the same interval, as given in the next column, must be added. We shall thus find, for example, that at the end of the 30th degree the piston has acquired precisely one-half of its velocity.

Third Column.

The acceleration of the motion of the piston varies as the cosine of the angle.

This column (c) gives the differences of the mean velocities, and shows how much the velocity is increased during each successive degree, or tenth of a degree.

The inspection of it is calculated, when made for the first

time, to fill the mind with astonishment. The same analysis when applied to the motion of a falling body, exhibits, as we have seen in Table No. XII., a uniform rate of acceleration; but here an acceleration appears, which is greatest at the absolute commencement of the stroke, and thence diminishes, at first very slowly, then more and more rapidly, all the way to the mid-stroke, when it insensibly becomes 0.

During the first two degrees the rate approximates very closely to uniformity, and so the acceleration at the commencement of the stroke is taken, in one Table, at twice the mean velocity of the piston during the first degree, and in the other Table at twice its mean velocity during the first one-tenth of a degree; but it will be seen that the latter is slightly the greater, and for the first one-hundredth of a degree, or 36″, it is, though almost imperceptibly, greater still. These are here shown together as follows:—

Twice the distance moved through during the first degree	·0003046096
Two hundred times the distance moved through during the first ·1 of a degree . .	·0003046173
Twenty thousand times the distance moved through during the first ·01 of a degree .	.0003046174

log =
-4.4837452

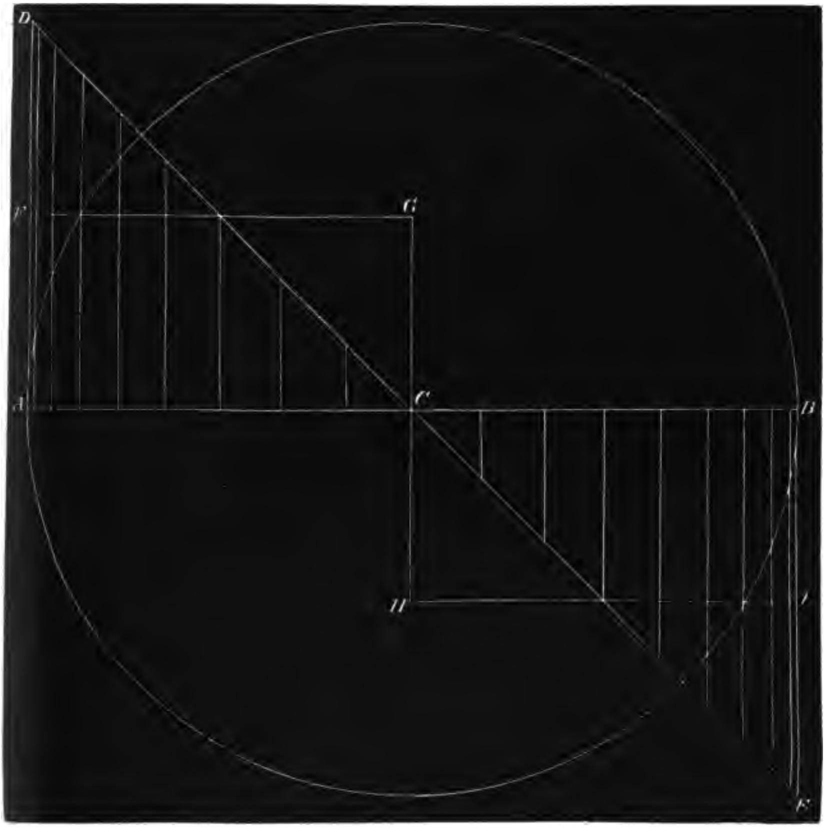
The distance moved through by a body uniformly accelerated increases as the square of the time. If, therefore, the above decimals were the same, the acceleration would be constant. The last column (*e*) in Table No. XIII. shows the nearly constant quantity by which the increase in the acceleration diminishes, as, tracing it backward through the first three degrees, we approach the line of centres.

One-tenth of a degree, when, for example, an engine is making four revolutions per second, is an exceedingly minute division of time, being $\frac{1}{14 \cdot 400}$ of a second. It is interesting to observe still the operation of the law of uniform acceleration; the distance moved through during a unit of time being one-half the velocity per such unit acquired. The Table showing the motion of the piston for each one-tenth of the first degree compares very well with Table No. XI. of the motion of falling bodies. The motion for ten degrees does not compare so well as during this time the diminution of the acceleration becomes considerable.

Now if we divide the cosine of any angle by the acceleration given in this column for that degree, we obtain always the quotient 3282·885, which expresses the constant ratio that the acceleration bears to the cosine. This number, it may be observed, is the exact square of the number 57·2965, which

lg = 3.5162462 (lg = 1.7581281

No. 45.



expresses the constant ratio of the velocity to the sine. The acceleration varies then directly as the cosines of the angles formed by the crank with the line of centres, and so inversely as the versed sines, or the distances traversed. Thus, for example, at the end of the 60th degree the piston has accom-

plished one-half the distance to the mid-point of its stroke, and has lost one-half its acceleration.

The retardation during the latter half of the stroke is, of course, the reverse of the acceleration during the former half.

The preceding diagram, No. 45, represents this acceleration, and subsequent retardation :—

Let A B represent the stroke of the piston, and A D the acceleration of its motion at the commencement of the stroke. Then will the work done in imparting its motion be represented by the right-angled triangle D A C, and that done in arresting it by the equal and opposite triangle C B E; for, if from any points on the line A B ordinates be drawn perpendicular to it, and of such lengths as to represent the acceleration or retardation at those points in the piston's stroke, or as to maintain a constant proportion with the lengths of the cosines, measured from such points to the mid-stroke C, then the terminations of all these ordinates will be found in the diagonal line D C E. The rectangles, A F G C and C H I B, equal in area to the above triangles, represent the equal work done by a constant acceleration and retardation, which would, within the same distance, impart and arrest the same velocity. The actual initial acceleration and terminal retardation are, it will be seen, twice those which would need to be constantly produced. Further demonstrations of this important action will be given presently.*

Fourth and Fifth Columns.

In these interesting columns, (*d*) and (*e*), are seen the increasing ratio in which the acceleration diminishes, and the manner in which, as the crank arrives at the end of the quadrant, it insensibly becomes 0; and also the diminishing ratio in which the diminution of acceleration increases. It is of course understood, that in the last column the unavoidable error in the last place of decimals will become apparent. The diminution of acceleration, column (*d*), maintains a constant ratio with the velocity, column (*b*), or with the sines of the angles; and the increase in this diminution, column (*e*), maintains a constant

* This increasing ratio of retardation explains the phenomenon of the water-hammer, or the concussion that is produced when a column of water is brought to rest by the resistance of a plunger, the action of which is controlled by a crank.

ratio with the acceleration, column (*c*), or with the cosines of the angles.

Before proceeding further it is important that we get a clear conception of the difference between velocity and acceleration. The acceleration of the piston's motion we have seen to be greatest on the very dead centre, where it has no velocity, where its motion begins; to diminish as the latter increases; and to become 0 at the mid-stroke, where the velocity reaches its maximum.

Now it is acceleration which requires the exertion of force to produce it. No force is, properly speaking, required to maintain motion once imparted. Force is then applied only to overcome resistance which would destroy the motion. Unobstructed motion continues for ever. But to give motion to a body, or take it away, to change its state from one of rest to one of motion, or from one degree of motion to another, or from one of motion to one of rest, this requires the exertion of force. So it is not when the piston is moving most rapidly, with a velocity already acquired, but when motion is being most rapidly imparted to it or abstracted from it, that the greatest force is being exerted to overcome its inertia; and this we have seen, and shall see still more clearly as we proceed, to be at the very commencement and termination of its stroke, when the crank is *on* the line of centres.

This action has, at the request of the writer, been investigated by an eminent physicist, Dr. F. A. P. Barnard, President of Columbia College in the city of New York. His paper on the subject, read before the Polytechnic Association of the American Institute, is published in their 'Transactions' for the year 1871–2. From it the following demonstration is taken:*—

"Supposing a piston to start from rest at the beginning of its course, and the crank to be maintained in uniform angular motion by some independent regulator, what must be the law of force acting on the piston so that it may complete the first half-stroke, without exerting any strain upon the crank, in the way either of acceleration or of retardation?

"Taking ϕ to represent the arc of revolution measured from

* This demonstration supposes the connecting-rod to be always parallel to itself, or to the axis of the cylinder, as it is assumed to be throughout this section.

zero at the line of centres, the differences of the versed sines of ϕ for equal successive minute intervals of time will be proportional to the velocities of the piston in such successive intervals. And the differences of these differences will be proportional to the successive accelerating forces required, in order that the uniformity of revolution may be maintained. These conditions may be best expressed in the notation of the differential calculus.

"Put, therefore,

ϕ = the arc of revolution to radius 1.
s = the space passed over by the piston.
r = the length of the crank.
t = time; v = velocity; f = accelerating force.
F = constant value of f at maximum.
T = constant time of revolution.
V = constant angular velocity of revolution.

"Then

$$V = \frac{d\phi}{dt} = \frac{2\pi}{T}. \qquad \frac{d\phi^2}{dt^2} = \frac{(2\pi)^2}{T^2}.$$

$$ds = r\,d\,(\text{v. s.}\,\phi) = r\,d\,(-\cos.\phi) = r\sin\phi\,d\,\phi.$$

$$v = \frac{ds}{dt} = \frac{r\sin\phi\,d\phi}{dt}. \quad \frac{dv}{dt} = \frac{r\,d(\sin\phi\,d\phi)}{dt^2} = \frac{r\cos.\phi\,d\phi^2}{dt^2}$$

Substituting for

$$\frac{d\phi^2}{dt^2} \quad \frac{dv}{dt} = \frac{(2\pi)^2}{T^2}\,r\cos.\phi.$$

But

$$\frac{dv}{dt} = f \mathrel{\therefore} f = \frac{(2\pi)^2}{T^2}\,r\cos.\phi.$$

"When $\cos.\phi = 1$ or -1, $f = \pm\dfrac{(2\pi)^2}{T^2}r = \pm V^2 r$ = centrifugal force of a unit mass of matter revolving in a circle of which r is the radius, the time of revolution is T. The accelerating force f, therefore, varies as $\cos.\phi$, and is maximum when $\cos.\phi$ is maximum. Hence,

$$F = \frac{(2\pi)^2}{T^2}r, \text{ and } \frac{F}{g} = \frac{(2\pi)^2 r}{32\cdot166\,T^2} = \frac{\pi^2 r}{8\cdot0416\,T^2}$$

which expresses the ratio of F to gravity.

"In order to compare this force with the constant force which would generate the same velocity in the same time, we observe that, by hypothesis, this force acts only during a quarter revolution of the crank, or a half stroke of the piston; and, hence, that the maximum velocity is necessarily equal to the uniform velocity of the extremity of the crank: or, in other words, since

$$v = \frac{r \sin\phi \, d\phi}{dt}; \text{ and } \frac{d\phi}{dt} = \frac{2\pi}{T}; \therefore v = \sin\phi \frac{2\pi r}{T}$$

which is maximum when $\sin\phi$ is maximum, or when $\phi = 90°$. Thence, putting V^1 for this maximum velocity,

$$V_1 = \frac{2\pi r}{T}$$

This velocity is generated in the time $\frac{1}{4}T$. Supposing it generated by a constant force, this force would be represented by the velocity it is capable of generating in one second; or, putting F_1 to stand for this constant force,

$$F_1 = \frac{V_1}{\frac{1}{4}T} = \frac{2\pi r}{\frac{1}{4}T^2} = 4\frac{(2\pi r)}{T^2}.$$

"From this we obtain

$$\frac{F}{F_1} = \frac{(2\pi)^2 r}{4(2\pi r)} = \frac{2\pi}{4} = \frac{1}{2}\pi.$$

Also

$$\frac{F_1}{g} = \frac{4(2\pi r)}{g\,T^2} = \frac{8\pi r}{32 \cdot 166\,T^2} = \frac{\pi r}{4 \cdot 0208\,T^2}.$$

If we take, for r and T, $r = 1 \cdot 25$ feet and $T = \frac{60}{122 \cdot 8}$ sec., we shall obtain, by substitution,

$$F = 205 \cdot 031; \text{ and } F_1 = 130 \cdot 527;$$

whence

$$\frac{F}{F_1} = 1 \cdot 57078 = \frac{1}{2}\pi \text{ as above.}$$

"This force F_1, however, is a greater force than would be required to generate the velocity, V_1, in acting constantly

through the *space* represented by r, if *time* is left out of the question; for, remembering that the spaces passed through under the influence of constant forces are governed by the law,

$$S = \tfrac{1}{2} f t^2$$

and applying this to the case in hand, in which the time to be considered is $\frac{1}{4}$T, we have

$$S = \tfrac{1}{2} F_1 (\tfrac{1}{4}T)^2 = \frac{8 \pi r}{T^2} \times \frac{1}{32} T^2 = \tfrac{1}{4} \pi r,$$

which is less than the radius r.

"Where different velocities are generated by forces acting through the same space, these forces are proportional to the squares of the velocities generated. And the velocities generated by the same force acting through different spaces are as the square roots of the spaces.

"Thus, gravity will generate, in acting through the space r,

the velocity $= g\sqrt{\dfrac{r}{\frac{1}{2}g}} = g\sqrt{\dfrac{2r}{g}} = \sqrt{2rg}.$

Hence, the force $F_{\prime\prime}$ which, in acting through the same space r, would generate the velocity $\frac{2\pi r}{T}$, will be found by the proportion,

$$\left(\sqrt{2rg}\right)^2 : \left(\frac{2\pi r}{T}\right)^2 :: g : \frac{(2\pi)^2}{2T^2} r$$

which is one-half the initial accelerating force F.

"Hence the force required to give to the piston the requisite initial acceleration is double that which, by acting constantly through a space equal to the length of the crank, would impart to the piston the same final velocity."

In spite, however, of these demonstrations, the proposition seems, in the absence of further explanation, quite incredible. We have been taught and accustomed to regard the piston as wholly passive at the terminations of the strokes; and the idea of a force and a resistance exerted in counteraction there, other than those involved in work done by the engine, is one exceedingly difficult to entertain. Moreover, fig. 45, representing the acceleration and retardation through the stroke,

shows an abruptness of beginning and ending which seems entirely opposed to what we know to be the manner in which the piston approaches and leaves the end of its stroke. Evidently we have not got to the bottom of the matter yet.

The explanation of the action of a piston during an entire revolution of the crank, to which attention is now invited, will remove this difficulty, which arises from that partial apprehension of the truth that cannot satisfy the demands of thoughtful and logical minds.

Let an engine be supposed from which the cylinder-heads have been removed, so that the motion of the piston may not be impeded by confined air, while it is caused to make its usual reciprocations by means of a band driving the crank-shaft with a uniform motion; and while it is thus being driven, let us note the resistance which the piston offers to the alternate accelerations and retardations that the crank produces in its motion. These resistances are exhibited in the following diagram, No. 46, as they are opposed to the crank at each point of its revolution.

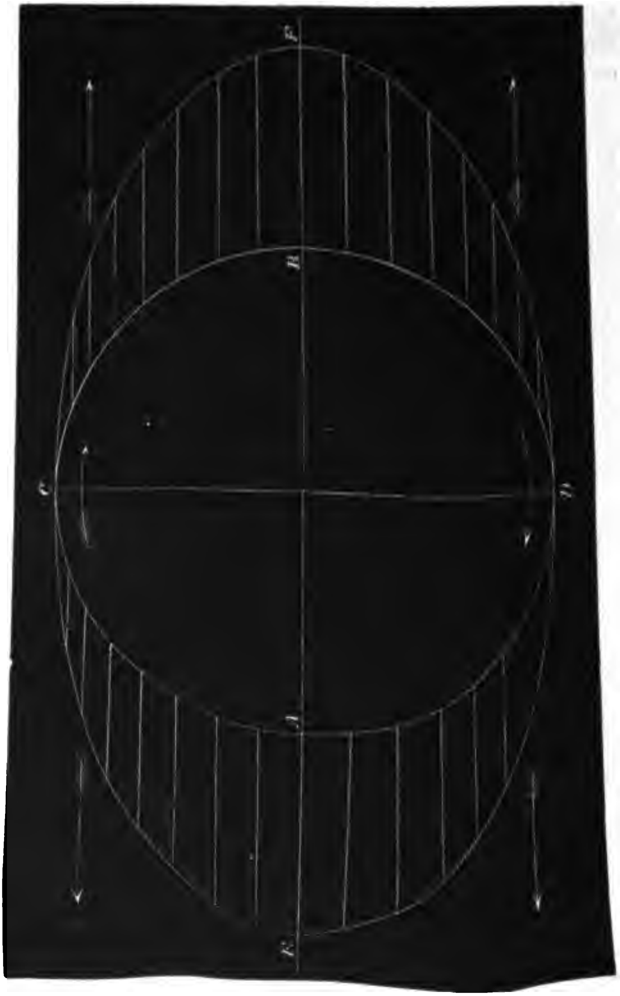
Let A C B D represent the circle described by the crank, E F the line of centres, and A E and B F the acceleration of the piston at the beginning, and its retardation at the termination of each stroke. Then the circumscribed ellipse will represent the forces required to produce the alternate acceleration and retardation throughout the revolution. The figure is the same as No. 45, the triangles are only bent around a circle, and the ordinates, which were vertical, have become horizontal.

At C the piston is moving with a velocity equal to that of the extremity of the crank,* and its motion begins insensibly to be retarded by the latter. It is immaterial in which direction the crank is rotating; but let it be supposed to be moving now in the direction shown by the curved arrows. The retardation increases in amount, as represented, up to the point B, where the piston is brought to rest, and begins to be impelled in the reverse direction. The impelling force thence diminishes, as shown, up to the point D, where it ceases, and retardation begins, culminating at A; at which point the piston, being

* The extremity of the crank is found at the centre of the crank-pin, the length of the crank being measured from the centre of the shaft to this point.

again brought to rest, begins to be impelled in the direction in which we first observed it moving with its full velocity at C.

Now there are here two things to be noted: first, the direction



No. 46.

in which the resistance of the piston to the impelling and retarding forces is exerted must be reversed twice in each revolu-

tion; and, second, this change does not take place on the dead centres, or at the same time with the change in the direction of its motion. It occurs at the middle point of each stroke, at the points C and D in the figure, where acceleration passes into retardation, while the motion continues in the same direction, and at the instant of the change the motions of the piston and the crank coïncide both in direction and velocity.

The straight arrows show the direction in which the resistance of the piston is being exerted, namely, from C to D in the direction B F, and from D to C in the direction A E. This will become clear on consideration. At C the piston begins insensibly to oppose its living force to the retardation of the crank. and resists it more and more forcibly up to the point B. At this point, on the dead centre, what takes place? The crank is now moving at right angles with the line of motion of the piston, and the direction of the piston's motion is insensibly reversed. Retardation of this motion in the direction B F passes, at its maximum, into acceleration of it in the reverse direction, at *its* maximum; but the resistance of the piston is in one direction, B F, all the time; the effort of the crank is in the opposite direction, F B, all the time: nothing is changed but only the direction of motion of the piston, as the force which has brought it to rest, continuing to act, impels it to return on its path. Each strain of the piston on the crank, alternately in opposite directions, begins insensibly at the mid-stroke.

This resistance of the piston, at its culminating point on the line of centres, is equal to the centrifugal force which the same mass would constantly exert if it were revolving at the extremity of the crank. This is proved as follows:—

Let us suppose a crank 1 foot in length to make, with uniform motion, 1 revolution in a minute: what would be the accelerating force exerted on the piston at the commencement of each stroke?

The motion of the piston may be regarded as being uniformly accelerated during the first degree, since the error involved in this assumption is too small to appear in our calculation, and this constant acceleration may be conceived to be continued indefinitely. We will suppose it to be continued through $6^\circ = 1$ second of time. Through what distance would the piston move? Answer: $\cdot 0001523048 \times 6^2 = \cdot 0054829728$ of a foot. An accelerating force equal to its own weight would move it in one second $16 \cdot 083$ feet. The accelerating force at

the commencement of the stroke must therefore be, in terms of the weight of the body,

$$\frac{\cdot 0054829728}{16 \cdot 083} = \cdot 000341;$$

which is the well-known coefficient of centrifugal force.

But the resistance of the piston *at this point* is not merely equal to, it *is* centrifugal force. What is this force? As exerted by a revolving body, it is the resistance which such a body opposes to being deflected from a direct line of motion; or, since relatively to the radial line the revolving body is at rest, its line of motion being tangential, or at right angles with the radius, it is the resistance which it opposes to being moved towards the centre from a state of rest, and the amount of its deflection, or motion along the radial line towards the centre, is the versed sine of the angle. So also the piston is, at this point, being moved towards the centre from a state of rest, the amount of its motion is the versed sine of the angle, and its resistance is exerted in the radial direction; therefore in its nature, as well as its amount, it is identical with centrifugal force.

When, therefore, we say that on each dead centre the piston is at rest, we mean, or we should mean, that it is at rest in the same sense in which a revolving body is always at rest relatively to the line connecting it with the centre. The true conception of it at these points is as exerting a radial strain on the crank, which varies according to the laws of centrifugal force: namely, first, directly as the mass; second, with a given length of crank as the square of the speed; third, with a given number of revolutions per minute, directly as the length of the crank; and fourth, with a given speed, inversely as the length of the crank.

At every other point in its stroke the resistance which the inertia of the piston offers to the accelerating or retarding force is the horizontal component of the centrifugal force which it exerts on the line of centres. The resolution into its rectangular components of the centrifugal force of a body revolving in a vertical plane, shows the cosine of the angle to be everywhere the coefficient of its horizontal component. But this we have seen, page 203, to be also the coefficient of the accelerating and retarding forces.

The action which we have now investigated receives its practical demonstration in the fact, that a horizontal engine is

perfectly balanced, in the horizontal direction, by a counterweight, equal in weight to the entire mass of the reciprocating parts, revolving opposite to the crank, and having its centre of gravity at a distance from the centre equal to the length of the crank. The vertical action of such a counterweight is resisted by the earth, and an engine so constructed may be run at any speed without disturbance of its stability.

We are now prepared to obtain a formula for the amount of the accelerating force at the commencement of the stroke. We have found already that the accelerating force required to impart its initial velocity to a piston weighing one pound, when the crank is 1 foot in length and makes 1 revolution per minute, is ·000341 of a pound.

Let, then,

W = the weight of the transmitting parts;
L = the length of the crank, in feet;
R = the number of revolutions per minute;
n = the constant number ·000341;
a = the area of the piston in square inches;
and p = the pressure on the square inch required.

Then will $R^2 Ln$ = the initial accelerating force, in terms of the weight of the mass, or the ratio which this force bears to the force of gravity; and $p = \frac{R^2WLn}{a}$

Example.—What is the force required, at the beginning of the stroke, to put in motion the piston of an engine of 30 inches diameter of cylinder and 4 feet stroke, making 60 revolutions per minute, and the reciprocating parts of which weigh 2000 lbs.?

$W = 2000$; $R^2 = 3600$; $L = 2$; $n = \cdot000341$; $a = 707$; and $p = 6\cdot945$ lbs. pressure on each 1 square inch of piston area.

This force varying as the square of the speed, if the speed of this engine were reduced to 40 revolutions per minute, it would be only 3·086 lbs.; and on the other hand, if the speed should be increased to 80 revolutions per minute, it would be 12·344 lbs. on each 1 square inch of piston area.

But how does this calculation show the initial accelerating force to compare with the constant force which would impart

the velocity attained at the mid-stroke? To answer this question, let the formula for a constant accelerating force, pages 205–6, be applied to the above example.

Substituting the values, we have,

$$f = \frac{2000 \times 157 \cdot 9144}{128 \cdot 664} = 2 \cdot 455 \div 707 = 3 \cdot 472 \text{ lbs.}$$

on each 1 square inch of piston, which is one-half of the actual initial force as found above.

Again, let the above formula for the initial accelerating force be applied to the case of the engine of short stroke and with a heavy piston, running at higher speed, employed for illustration on pages 205–6.

Then we have, $W = 1100$; $R^2 = 19{,}600$; $L = 1 \cdot 25$; $n = \cdot 000341$; $a = 201$; and $p = 46$ lbs. on each 1 square inch of piston: so that again the initial force equals twice the constant force that would impart the same final velocity.

Again; what must be the aggregate weight of the reciprocating parts of an engine of 18" bore, and 12" stroke, making 400 revolutions per minute, in order that the initial accelerating force shall be equal to 60 lbs. on each 1 square inch of piston?

$$R^2 L n = 27 \cdot 28 \text{; and } \frac{254 \cdot 5 \times 60}{27 \cdot 28} = 560 \text{ lbs.}$$

By this simple calculation the initial accelerating force may in all cases be ascertained, and the triangles described, representing the work first absorbed and then imparted by the piston, and which must be subtracted from the first half and added to the latter half of the diagram, in order to show the real distribution of pressure on the crank.

In the following diagram, No. 47, this action and its practical value are illustrated. The steam pressure is taken at 90 lbs., with a vacuum of 12 lbs., making the total initial pressure 102 lbs. on the square inch, and the point of cut-off is taken at one-sixth of the stroke. For plotting the expansion curve, 4 per cent. is added to the length of the stroke for the waste room in clearance and passages.

Two cases are represented—one in which the initial pressure required to overcome the inertia of the piston is 8 lbs. on the square inch, and which fully represents the requirements for this purpose in the best engines of the usual speed and

length of stroke, in the large majority of which the force thus absorbed is much less than this—and the other in which the initial pressure absorbed in this manner is 46 lbs. on the square inch of piston, as in the engine last above described.

The figure C E F L D, representing the preponderance of the

No. 47.

steam pressure on this side of the piston during the stroke, shows how excessively unequal the pressures applied to the crank at the opposite ends of the stroke would be in engines cutting off at this point, if the reciprocating parts were with-

out weight. The figure I E F L K shows in how slight a degree this inequality is corrected in the best ordinary practice. The G E F L H shows the result obtained by increasing the inertia of the reciprocating parts in a judicious manner, with a view to correcting this lamentable defect in engines working steam expansively, as all economical engines must do.

Although we are not yet arrived at the point, from which the beneficial effect of the action thus obtained can be fully seen, still the inspection of this figure makes it obvious that, by means of it, first, the crank is relieved of nearly one-half of the initial force of the steam; second, the pressures exerted on the crank at the opposite ends of the stroke are nearly equal; and third, the amount of pressure on the crank is least during that portion of the stroke where the rotative effect of a given pressure is greatest. There is obviously no loss of power in this action. The transmitting parts act merely as a reservoir of power, like the fly-wheel, and impart all that they receive.

Illustrations of this action of the inertia of the reciprocating parts of an engine in modifying the distribution of the pressure on the crank might be multiplied. Engineers can readily ascertain that which takes place in their own practice. The single example here introduced illustrates the action in cases in which steam is worked to the best advantage.

SECTION III.

OF THE MODIFICATION OF THE ACCELERATION AND RETARDATION OF THE PISTON THAT IS OCCASIONED BY THE ANGULAR VIBRATION OF THE CONNECTING-ROD.

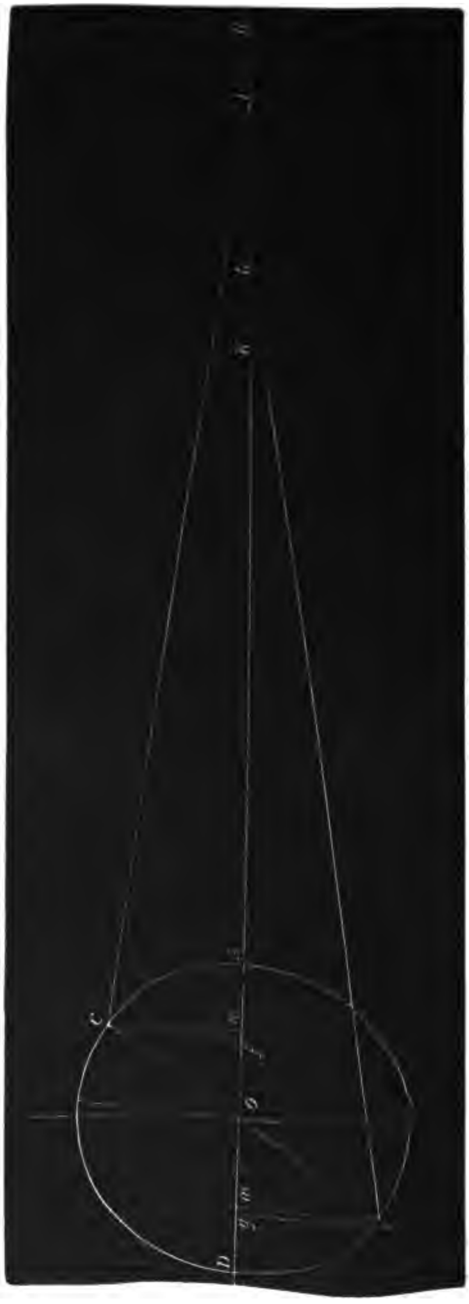
In the preceding investigation the angular vibration of the connecting-rod has, for the sake of clearness, been disregarded, and the subject has been considered as if no such action took place. In practice, however, this vibration always does take place, and it produces a considerable difference in the velocity of the piston in the opposite ends of the cylinder, and in the force required to impart and arrest it.

The usual proportionate lengths of connecting-rods are 4, 5, and 6 times the length of the crank. They are rarely employed longer than the last, and still more rarely shorter than the first, of these lengths. The greater their proportionate lengths, the less, obviously, is their angular vibration during a revolution of the crank.

The vibration of the connecting-rod operates to increase the retardation and acceleration of the piston in the end of the cylinder farthest from the crank in direct-acting engines, and in the upper end of the cylinder in beam-engines, and to diminish them in the opposite end. No one can observe attentively the motions of a piston-rod or cross-head, without perceiving a very marked difference in the speed with which they approach and leave the opposite ends of the stroke.

The inequality thus occasioned is greatest at the extreme points of the stroke. The retardation and acceleration are more rapid on the forward* dead centre than on the back* one,

* In accordance with locomotive usage, the term "forward" is employed to designate the end of the cylinder farthest from the crank, and the stroke of the piston in that direction, and the terms "back" and "return" to designate the end of the cylinders nearest to, and the stroke of the piston towards, the crank.



No. 48.

by 40 per cent. when the connecting-rod is 6 cranks in length, 50 per cent. when it is 5 cranks in length, and 66 per cent. when it is 4 cranks in length. These enormous differences rapidly diminish, however, and at about the mid-stroke the difference of course disappears. The average difference between the velocities of the piston through the opposite half-strokes is one-half the difference between the velocities at the extremes, as above given.

The opposite diagram, No. 48, illustrates the manner in which the vibration of the connecting-rod operates to produce these inequalities:—

Let D H be the centre line of an engine, and A C D E the circle described by the extremity of the crank. C O E represents the crank in opposite positions, and A C and D E are equal arcs. H K represents the stroke of the piston. The lines I C and L E represent the connecting-rod, which is shown 6 cranks in length. A *n* and D *m* are equal, being the versed sines of the equal angles C O A and D O E; *n f* and *m g* are also equal, being the versed sines of the equal angles C I *f* and E L *g*. While the crank moves through the arc A C, the piston moves the distance H I, which is equal to A *n* + *n f*; and in the opposite quadrant, while the crank moves through the arc D E, the piston moves only the distance K L, which is equal to D *m* − *g m*.

This action may be expressed as follows:—

Let A = the angle formed by the crank with the line of centres;
B = the angle formed by the connecting-rod with the same;
C = the ratio between the length of the connecting-rod and that of the crank; and
D and D′ = the distances moved by the piston in the forward and in the back ends of the cylinder.

Then will D = versin A + (C versin B) and
D′ = versin A − (C versin B).

The three Tables at the end of the book have been computed by the above formula. These give, for each 1° of the entire revolution of the crank:—

First. In the outer column, the total distance moved by the piston from the commencement of each stroke.

Second. In the next column, the mean velocity of the piston while the crank is moving through each degree;

Third. In the third column, the acceleration or retardation of the piston at the end of each degree; and,

Fourth. In the inner column, the diminution of the acceleration, or increase of the retardation, during each 1°, in the three following cases, namely:—

When the length of the connecting-rod is equal to 6 times, 5 times, and 4 times the length of the crank, the length of the crank is taken as 1.

When the velocity of the piston at the *end* of any degree is required, then one-half the acceleration or retardation for that degree must be added to, or subtracted from, the mean velocity.

At the end of the 90th degree, the velocity of the piston is always equal to that of the extremity of the crank.

The velocity of the piston exceeds that of the crank, through 18° of the quadrants nearest to the cylinder when the connecting-rod equals 6 cranks, through 21° when the connecting-rod equals 5 cranks, and through 26° when the connecting-rod equals 4 cranks in length; and it reaches its maximum, and retardation begins, at about 81°, 79°, and 77° of the quadrants nearest to the cylinder, in the above three cases respectively.

The coefficients of distance and velocity given in these Tables may be employed as follows:

To ascertain the distance travelled by the piston, from the commencement of the stroke, when the crank has arrived at the end of any degree of its first quadrant, multiply the decimal for that degree in the outer column by the length of the crank; when it has arrived at the end of any degree of its second quadrant, multiply as above, and subtract from twice the length of the crank.

To ascertain the mean velocity of the piston while the crank is traversing any degree of its arc, multiply the decimal for that degree in the second column by the piston travel, and divide by the decimal ·011111, since the mean velocity of the piston, for its entire stroke, is to its final velocity as ·011111 is to ·0174532.

If the result comes larger than you expect, remember that at the mid-stroke the piston has a velocity greater than its travel or mean velocity, in the proportion of 1·5708 to 1. If the velocity at the *termination* of any degree is required, then one-half the acceleration for that degree must first be added, or one-half the retardation subtracted, as already explained.

In the following table are presented in one view the comparative velocities of the piston, at the expiration of each interval of 5° in the revolution of the crank, under the following conditions; namely, when the connecting rod is of infinite length, and when it is six times, five times, and four times the length of the crank. The first is the theoretical condition under which the angular vibration of the rod disappears. These velocities are seen in comparison with the velocity that the piston would acquire if its initial rate of acceleration, with a connecting-rod of infinite length, could be constantly maintained.

TABLE XIV.

Showing the Velocity of the Piston at the termination of each interval of 5° of the revolution of the crank, counting from the line of centres—

FIRST, *under the theoretical condition of a connecting-rod of infinite length, when its angular vibration disappears; and,*

SECOND, *under three practical conditions, namely, when the length of the connecting-rod is* SIX *times, when it is* FIVE *times, and when it is* FOUR *times the length of the crank—*

in comparison with the velocity that would be imparted to it by constant acceleration under the first of the foregoing conditions; that so imparted at the termination of the first degree being taken as the unit.

The initial acceleration, represented by twice the mean velocity during the first degree, is, in the above four cases respectively, as follows:—

1. With con.-rod of infinite length	·0003046096, which if we take to		= 1, then
2. With „ = to 6 cranks	·0003553790, at forward end of cylinder, will		= 1·166
	·0002538402, at back	„ „ „	= ·833

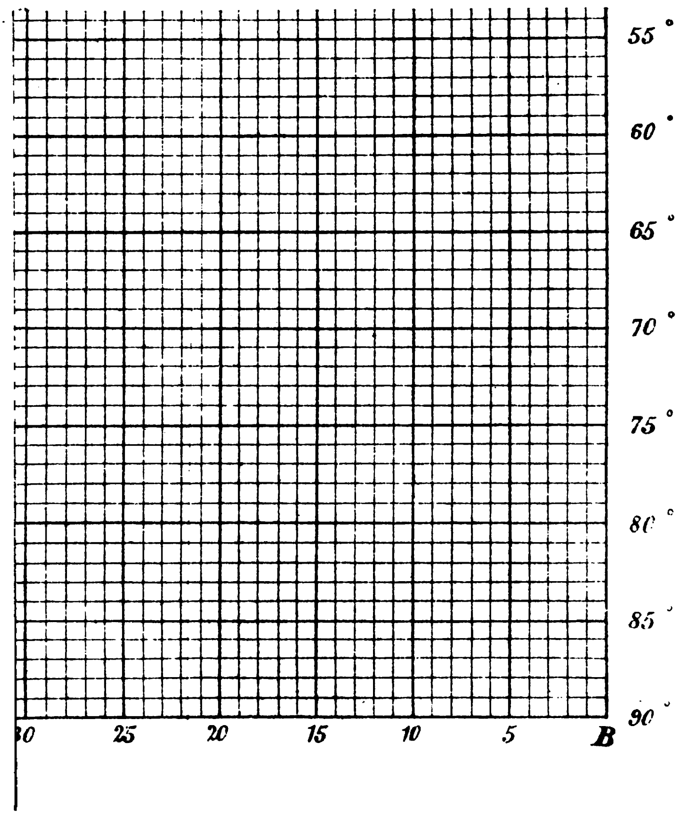
3. With con.-rod = to 5 cranks { ·0003655326, at forward end of cylinder = 1·2
·0002436866, at back " " " = ·8
4. With " = to 4 cranks { ·0003807638, at forward " " " = 1·25
·0002284554, at back " " " = ·75

Angular position of the crank.	1. Connecting-rod of Infinite Length.		2. Connecting-rod = to 6 Cranks.		3. Connecting-rod = to 5 Cranks.		4. Connecting-rod = to 4 Cranks.	
	Velocity that would be imparted by constant acceleration.	Velocity actually acquired.	Velocity acquired In forward end of cylinder.	Velocity acquired In back end of cylinder.	Velocity acquired In forward end of cylinder.	Velocity acquired In back end of cylinder.	Velocity acquired In forward end of cylinder.	Velocity acquired In back end of cylinder.
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)
5°	5	4·994	5·82	4·168	6·00	3·99	6·24	3·75
10	10	9·94	11·58	8·30	11·91	7·97	12·40	7·48
15	15	14·83	17·22	12·44	17·70	11·96	18·42	11·24
20	20	19·60	22·67	16·53	23·29	15·91	24·22	14·98
25	25	24·215	27·88	20·55	28·60	19·83	29·73	18·70
30	30	28·65	32·80	24·50	33·66	23·64	34·90	22·40
35	35	32·86	37·37	28·35	38·28	27·44	39·66	26·06
40	40	36·83	41·56	32·10	42·52	31·14	44·00	29·66
45	45	40·515	45·32	35·71	46·30	34·73	47·79	33·24
50	50	43·89	48·63	39·15	49·60	38·18	51·07	36·71
55	55	46·93	51·53	42·33	52·39	41·47	53·80	40·06
60	60	49·62	53·80	45·44	54·66	44·58	55·97	43·27
65	65	51·93	55·63	48·23	56·39	47·47	57·56	46·30
70	70	53·84	56·945	50·735	57·59	50·09	58·57	49·11
75	75	55·34	57·76	52·92	58·26	52·42	59·00	51·68
80	80	56·42	58·08	54·76	58·45	54·39	58·95	53·89
85	85	57·08	57·93	56·23	58·09	56·07	58·36	55·80
90°	90	57·30	57·30	57·30	57·30	57·30	57·30	57·30

This Table is represented in the lithographed diagram, No. 49, in which are also shown the acceleration and retardation of the piston, and the distance travelled by it, at every point in the revolution of the crank.

The height A B of the right-angled triangle A B C represents 90°, reckoned from the line of centres. The apex A represents the beginning and end of the stroke, and the base B C represents the point in the stroke at which the piston stands when the crank stands at 90°. This point would be the middle of the stroke if the connecting-rod were without angular vibration. Its actual position is seen in the Tables at the end of the book. The inclination of the diagonal A C represents a constant acce-

... of the velocity to the sine
3282.885 (or 57.3²) is the ratio of acceleration to cosine


55°
60°
65°
70°
75°
80°
85°
90°
30
25
20
15
10
5
B

leration of the piston, the initial acceleration that it would have with a connecting-rod of infinite length being conceived to be uniformly maintained. The base B C, and all lines parallel with it, and terminating on the vertical A B and the diagonal A C, represent the velocity at every point which would be imparted by such constant acceleration. Those at the end of each 5° correspond to column (*a*) in the Table. The area of the triangle above any horizontal line represents the total distance that a piston thus uniformly accelerated would travel in acquiring the velocity represented by such line.

We will now confine our attention to the first quadrant. The central curve represents at each point, by its inclination from the vertical, the acceleration that would be applied to the piston at that point in the quarter-revolution of the crank, if the connecting-rod were without angular vibration. The horizontal lines, drawn to it from the vertical A B, represent the velocity that it would have at those points, imparted by such diminishing acceleration, and those at the end of each 5° correspond to column (*b*) in the Table. The included areas represent the distances through which the piston would travel in acquiring these velocities.

The divergence of the curves A and A′ from the central curve exhibits the effect of the angular vibration of a connecting-rod the length of which is 4 times that of the crank, and that of the curves B and B′ exhibits this effect when the length of the rod is 6 times that of the crank. The curves showing the effect of the vibration of a rod 5 cranks in length would fall between the curves A and B, and A′ and B′, and have been omitted for the sake of clearness.

The curves A and B represent the acceleration of the piston during the quarter-revolution of the crank on its forward stroke or in the end of the cylinder nearest to the crank, and the curves A′ and B′ represent its acceleration and beginning of its retardation during the same period on its return stroke. The horizontal lines, measured from the vertical A B to these curves respectively, show the velocity acquired by the piston at each point, and at the end of each 5° correspond to columns (*c*), (*d*) (*g*), and (*h*) of the Table.

These curves may properly be conceived to return upon themselves, so as to represent the alternate accelerations and retardations of the piston's motion during an entire revolution of the crank. The central curve returns upon itself continually, the

acceleration on each stroke being the same, and the retardation its reverse, and the mid-stroke coinciding with 90°. The other pairs of curves, however, return, each on itself, at the beginning and ending A, but on each other at the mid-stroke E, because the acceleration and retardation are the reverse of each other in the same end of the cylinder, but not so at opposite ends of a stroke.

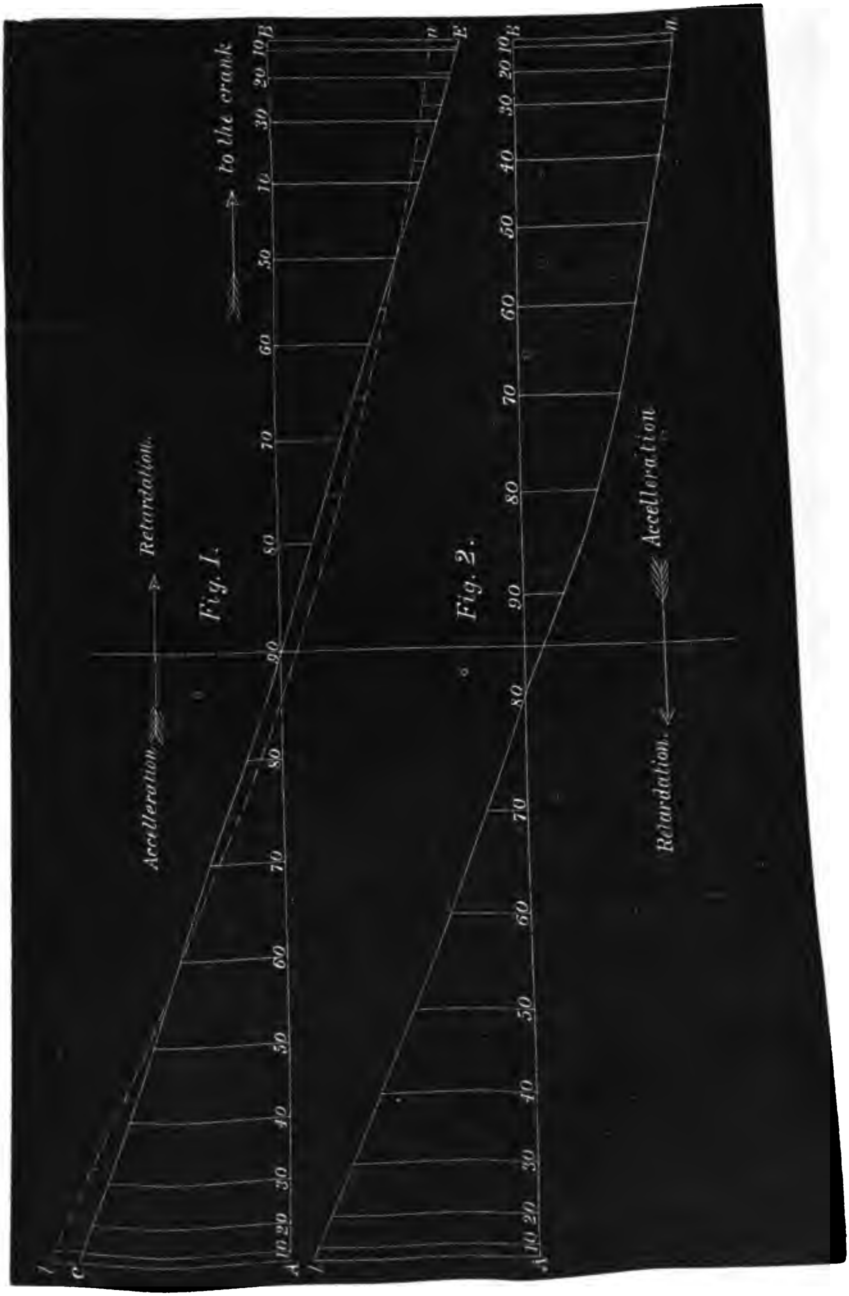
The effect of the angular vibration of the connecting-rod, in increasing or diminishing the velocity of the piston on its opposite strokes, is greatest when the crank is at the angle of 45°. In the Tables at the end of the book the sines are drawn from the line of centres, and the cosines from the angle of 90°, to this culminating point.

The distinction between velocity and acceleration must be kept clearly in mind. We have before observed how opposite these are to each other; another illustration is seen just here. The angle of 45°, at which the angular vibration of the rod produces the greatest change in the velocity, is about the point at which, as will be seen in Table XV., this vibration does not affect the acceleration or retardation at all. Velocity has been incidentally considered here as part of the general subject, but in this section we shall have no more to do with it.

The following diagram, No. 56, represents the force required to effect the alternate acceleration and retardation of the piston, through the entire revolution of the crank, when the connecting-rod is 6 cranks in length. The lines A B in the two figures represent the stroke of the piston, and the vertical line shows the middle point of the stroke. The ordinates represent, at intervals of 10° of arc, the position of the piston in its stroke, when the crank is at the corresponding points of its semi-revolution.

The diagonal C E in the upper figure represents the acceleration and retardation as they would take place were there no angular vibration; and the dotted curve shows them as they are, with rods of this proportionate length. The curve in the lower figure is a duplicate of the dotted curve in the upper one. The positions of the ordinates in the lower figure, corresponding with the dotted ordinates in the upper one, show how much farther from the forward end of the cylinder, and how much nearer to the back end, the piston is, on account of the angular vibration of the rod, at the end of each 10° of the opposite quarter-revolutions of the crank. Acceleration passes into retardation at the point where the curve crosses the line A B

No. 56.


to the crank
Retardation.
Accelleration
Fig. 1.
Fig. 2.
Accelleration
Retardation.

The following Table has been prepared for practical use. It gives the coefficients of the accelerating or retarding forces for each 5° of the revolution, and for the three proportionate lengths of connecting-rod. The force required varies directly as the rate of acceleration or retardation, because merely the distance is increased or diminished, within which a given velocity is imparted or arrested.

TABLE XV.

Giving the force required to impart or to arrest the motion of the reciprocating parts of an engine, at the termination of each interval of 5° in the revolution of the crank, when the length of the connecting-rod is equal to 6 cranks, 5 cranks, and 4 cranks; the force so required on the dead centre, with connecting-rod of infinite length, being taken as 1.

Degrees.	Connecting-rod = 6 cranks.		Connecting-rod = 5 cranks.		Connecting-rod = 4 cranks.	
	Forward end of cylinder.	Back end of cylinder.	Forward end of cylinder.	Back end of cylinder.	Forward end of cylinder.	Back end of cylinder.
0°	1·1665	·8335	1·2000	·8000	1·2502	·7498
5	1·1605	·8322	1·1900	·8000	1·2433	·7495
10	1·1420	·8283	1·1730	·7968	1·2209	·7492
15	1·1110	·8211	1·1410	·7922	1·1835	·7482
20	1·0676	·8115	1·0942	·7853	1·1336	·7460
25	1·0148	·7981	1·0377	·7758	1·0705	·7426
30	·9508	·7817	·9678	·7638	·9947	·7370
35	·8780	·7610	·8900	·7482	·9091	·7292
40	·7968	·7361	·8030	·7292	·8139	·7180
45	·7088	·7058	·7091	·7052	·7114	·7029
50	·6149	·6710	·6093	·6760	·6021	·6830
55	·5171	·6300	·5062	·6412	·4900	·6572
60	·4166	·5834	·4000	·6000	·3749	·6251
65	·3148	·5305	·2935	·5519	·2603	·5847
70	·2134	·4708	·1871	·4970	·1467	·5371
75	·1130	·4048	·0827	·4347	·0364	·4809
80	·0154	·3322	— ·0180	·3650	— ·6830	·4156
85	— ·0791	·2538	— ·1139	·2886	— ·1668	·3414
90°	— ·1694	·1694	— ·2045	·2045	— ·2584	·2584

The initial accelerating force for any engine having been found by the formula given on page 227, this, multiplied into the proper column of coefficients given in the above Table, according to the proportionate length of the connecting-rod, gives a permanent Table for that engine, of the accelerating and retarding forces exerted, in pounds on the square inch of piston, at each 5° in the revolution of the crank.

This Table having been found, it may be applied, in the manner illustrated below, to all diagrams taken from that engine, and the resulting diagram will represent with absolute exactness, less the loss from friction, the pressure on the crank, at every point in its revolution.

We will take, for illustration of this method, diagram No. 47, on page 229, and will assume the initial accelerating force to be 46 lbs. on the square inch, as the mean of the two strokes, as it is represented in that diagram; and we have to find the correction necessary to be applied to the diagonal line G H.

The accelerating and retarding forces will be as follows, in pounds on the square inch of piston:—

n = 1100

Engine 16" diam cyl

12" stroke

R = 442

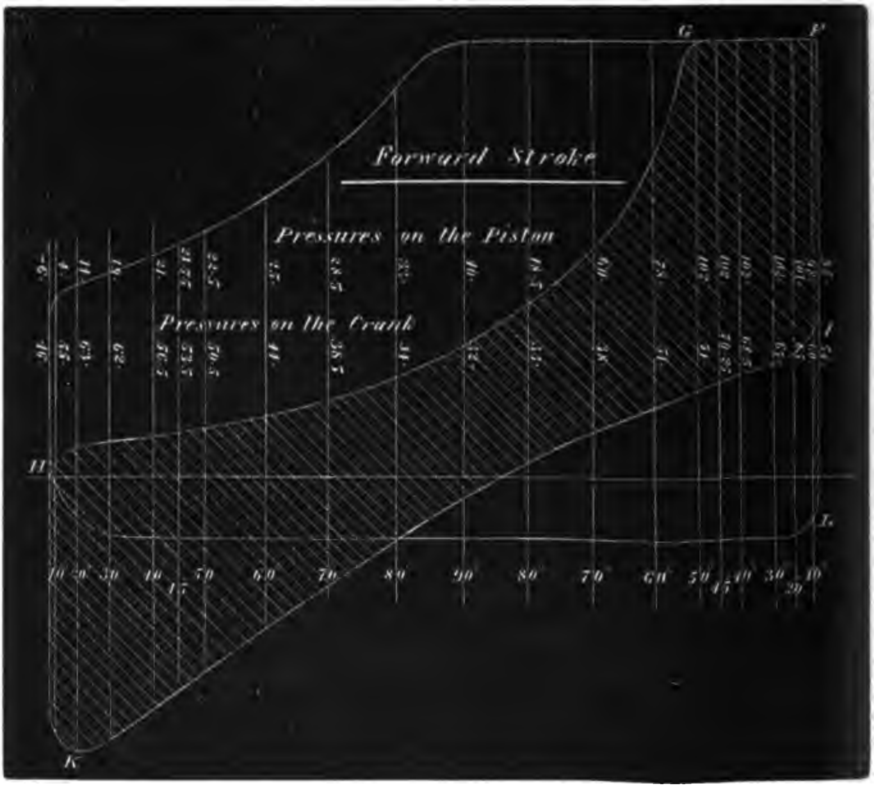
L = 1.25

Degrees.	In forward end of cylinder.	In back end of cylinder.
0°	53·659	38·341
10	52·532	38·102
20	49·110	37·329
30	43·737	35·958
40	36·653	33·861
45	32·605	32·467
50	28·285	30·866
60	19·164	26·836
70	9·816	21·661
80	·708	15·281
90°	7·792	7·792

We have next to determine the positions of the ordinates on which these pressures are to be measured, or the positions of the piston corresponding with the above positions of the crank. The usual method, which is to set off equal divisions on a semicircle, of which the atmospheric line of the diagram is the diameter, so that the intersections of the ordinates with the arc

will represent the positions of the crank, and with the atmospheric line the corresponding positions of the piston, would be correct if the inequalities we are considering did not exist, bu as engines are made at present, it is quite incorrect. To mark these points correctly the arc of revolution must be drawn separately, and from the divisions on this arc the points on the atmospheric line must be marked with compasses set to the proper length to represent the connecting-rod.

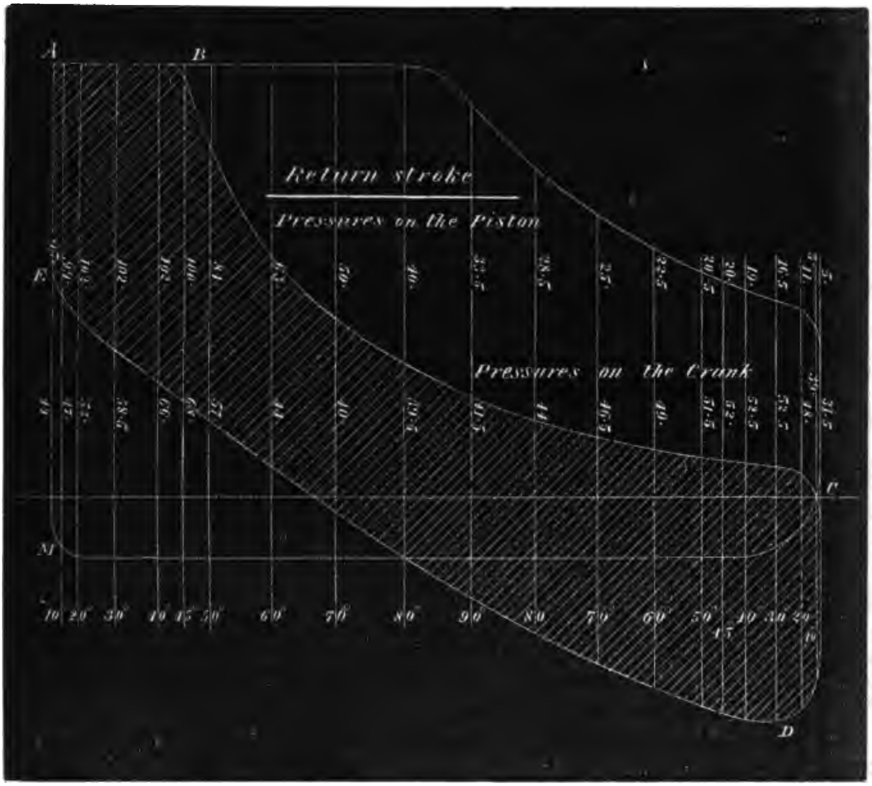
No. 57.



These two diagrams, No. 57 and No. 58, are copies of diagram No. 47, as these would be taken from the two opposite ends of the cylinder. The ordinates having been drawn in the manner just directed, the accelerating and retarding forces

above given being measured on them from the lines M C and L H, their extremities are found to fall into the curves E D and I K, and we have the figures A B C D E and F G H K I, representing the actual pressure on the crank at every point in its revolution. The pressures on the piston, and those on the crank, are marked on each ordinate.

No. 58.



It will be observed that the curves I K and E D are permanent. Whatever the pressure of steam, or the nature of its distribution, these remain unchanged. Also, if the steam be cut off considerably earlier, or if it follow considerably further than is here shown, this will make but little difference in the approximate equality of the pressures at the opposite ends of

the stroke. This, to one familiar with expansion curves, will be sufficiently obvious, but to give to it an extreme illustration the diagrams have been extended to show the steam cut off at half-stroke. The effect of the action of the transmitting parts is then seen in relieving the crank on the admission, and in maintaining the pressure during the latter part of the stroke. The contrast shows more clearly the especial and remarkable adaptation of this action to equalise the pressure in cases precisely as they exist in the best practice, namely, when the steam is cut off at from one-fourth to one-eighth of the stroke.

The subject of the distribution of the pressure on the crank has now been fully examined, and the data provided for determining this distribution in every case. In the illustrations employed the best practice has been shown, in respect both to the method of working steam, and the proportionate length of connecting-rod.

SECTION IV.

OF THE ROTATIVE EFFECT OF THE PRESSURE EXERTED ON THE CRANK.

We have considered the pressure exerted on the crank, by the combined action of the force of the steam and the inertia of the transmitting parts of the engine. We come now to inquire, what is the tangential value of this pressure, or its effect in producing rotation, at each point in the revolution of the crank?

We will suppose the crank to rotate, as it is commonly supposed to do, with a uniform velocity. The velocity of the piston, on the contrary, is continually changing. On the dead centres the piston has no velocity; at the end of 1° from the line of centres its velocity, as the mean of the opposite strokes, is to that of the crank as 1· is to 57·3; at the angle of 90° only the speeds of the two are the same, and the mean speed of the crank exceeds that of the piston in the proportion of 1·5708 to 1·, because it has to traverse one-half the circumference of a circle, while the piston is moving a distance only equal to its diameter.

We are already familiar with the law that force is the equivalent of distance. The dynamic value of a force acting through a given distance is the same as that of a greater force acting through a proportionately lesser distance, or of a lesser force acting through a proportionately greater distance. If the product of the force into the distance is the same, then the same work is done. To produce a given dynamic effect, the force must vary inversely as the distance through which it is exerted.

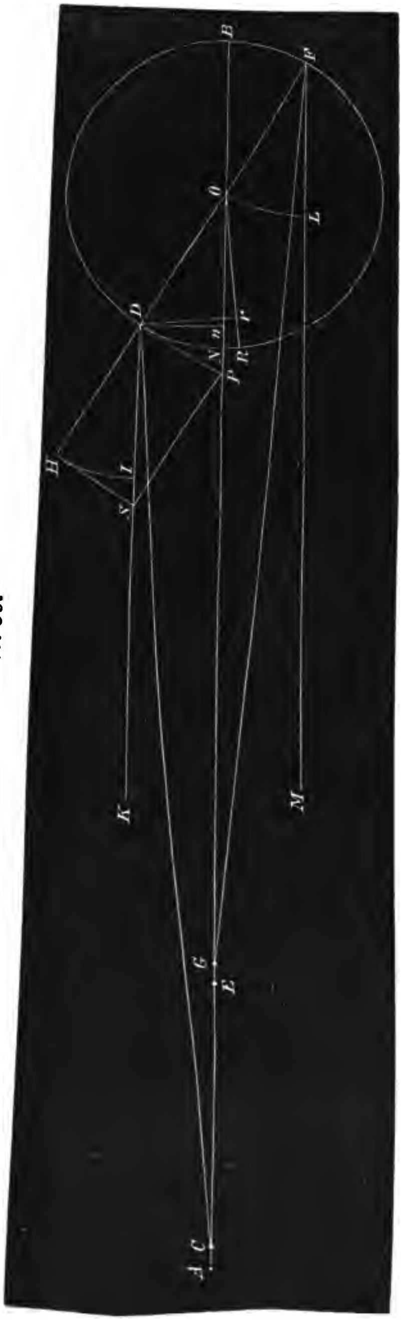
Let us conceive the transmitting parts of the engine to be without weight. The same work is being done at either end of the connecting-rod, and so the rotative effect on the crank of a constant pressure on the piston must vary precisely as the velocity of the piston varies. For, the pressure on the piston multiplied by its velocity is equal to the rotative effect exerted on the crank multiplied by *its* velocity. But the pressure on

the piston and the velocity of the crank are both, by supposition, constant. Then the other members of the equation—the velocity of the piston and the force acting to rotate the crank—vary in the same ratio.

If, therefore, the velocity of the crank, or of the piston when the crank is at 90°, be taken to represent the pressure on the crank at any other point of its revolution, then the velocity of the piston at this latter point, as given in the second column of the Tables at the end of the book, will express the tangential effect of the pressure at that point; or—disregarding the effect of the angular vibration of the connecting-rod—if 1· be taken to represent both the pressure on the crank and the velocity of the piston when the crank is at 90°, then the sine of the angle will, at every point in the revolution of the crank, express both the velocity of the piston and the rotative effect of the pressure at that point.

There are doubtless some who would be glad to examine this relation more closely, and for their use the following demonstration is given.

We will first suppose the pressure to be applied to the crank on lines parallel with the line of centres; as, for example, along the lines K D and F M in the following figure, No. 59. Then its rotative effect will vary as the sine of the angle which the crank forms with the line of centres. This is shown by resolving the force into its rectangular components. Let A B, in this figure, represent the centre line of an engine, and B F N D the circle described by the extremity of the crank. Let the distance S D represent the force applied along the line K D, to the crank at the point D. Draw the rectangle H S P D, of which the two sides H S and P D are parallel with the direction in which the crank is at that instant moving. Then either one of these sides will represent the effective component of the force, or the rotative effect produced. If S D is taken equal to P O, the secant of the angle which the crank forms with the line of centres, as is done in this figure, then D P will be the tangent. But the secant of an angle is to the radius as the tangent is to the sine. We may, therefore, substitute O D, the radius, in place of the secant O P, to represent the total force, when the sine D n will represent its effective component. The rotative effect of the pressure varies then as the sine of the angle. But we have seen, page 214, that under the condition here assumed, of infinite length of connecting-rod, the velocity of the piston varies as the



No. 59.

sine of the angle. Therefore the rotative effect varies directly as the velocity of the piston.

But the pressure is not applied to the crank on lines parallel with the line of centres, as is above supposed, but through the connecting-rod. We have seen how the angular vibration of this rod modifies the distribution of the pressure on the crank; we have now to inquire in what degree it modifies also the rotative effect of this pressure.

The vibration of the connecting-rod increases the angle at which the force is applied to the crank through the two quadrants nearest to the cylinder, and in the same degree diminishes this angle through the two quadrants farthest from it. This is shown in the figure in the preceding page, in which A G represents the stroke of the piston, O F and O D represent the crank in opposite positions, and C D and E F represent corresponding positions of the connecting-rod. The line F O D is produced to H, and K D and M F are drawn parallel with A B. Now by construction the angles H D I and O F L are equal; but the angle H D C is greater than H D I, and the angle O F E is less than O F L, and the angle added to H D I is equal to the angle subtracted from O F L.

How does this modify the tangential effect of the pressure? To find, for example, the effect of the pressure applied to the crank at the point D, along the line C D, draw O R parallel with C D. Then a resolution of the force, as before, will show its effective component to be represented by D r, the sine of the angle D O R. But by construction the angle N O R is equal to the angle O C D, as is also the angle M F E; whence it appears, that if we add to the angle formed by the crank with the line of centres for the half-revolution nearest to the cylinder, and subtract from this angle for the half-revolution farthest from it, the angle formed with the same line by the connecting-rod,— the sine of the sum, or of the difference, of these angles will be the co-efficient of the rotative effect of the force.

We say *it appears*, because this conclusion would not be correct. It would be correct if the end of the connecting-rod, at C for example, were moving along its own axis, in the direction C D; but such is not the case: it is moving instead along the line A G. This introduces a new element into the problem, which at first glance seems to render it obscure, but by the aid of the following diagram, No. 60, we shall readily ascertain the additional motion which the piston has on this

No. 60.

account, in both ends of the cylinder, and so the additional rotative effect of the force. The connecting-rod is here shown four cranks in length for greater clearness of illustration.

Let A B represent the line of centres, and C B D E the circle described by the revolution of the crank. The line C O D represents the crank in opposite positions, and G C and H D represent the corresponding positions of the connecting-rod. A and F mark the extremes of the stroke.

If the connecting-rod had moved from the commencement of either stroke continually along its axis, it would now have arrived on the forward stroke at the position D M, and on the return stroke at the position C P. But, moving along the line A F, the piston has on the forward stroke moved through F H, which is greater than F M by the distance *m* H, and on the return stroke it has moved through A G, which is greater than A P by the distance *n* G.

For minute angles, the arcs P G and M H will not differ sensibly from tangents of the arcs P *n* and M *m*, so that A G will be the secant of the arc P *n*, and F H the secant of the arc M *m*. The velocity of the piston, or the tangential effect of the pressure on the crank, will then be represented by the sine of the sum, or of the difference, of the two angles as above multiplied by the secant of the angle formed by the connecting-rod with the line of centres.

This is expressed in the following formula :—

Let A = the angle formed by the crank with the line of centres
B = „ „ con.-rod „ „

and let R and R′ = the rotative effects exerted in the quadrants nearest to, and farthest from, the cylinder respectively;

Then will $R = \sin (A + B) \times \sec B$

And $R' = \sin (A - B) \times \sec B.$

The following table has been computed by this formula. It gives, for intervals of 5° through the revolution of the crank, the coefficients of the rotative effect of the pressure, when the length of the connecting-rod equals 6, 5, and 4 cranks.

Concerning this Table it is to be observed that,—

1st. These coefficients agree with those for the velocity of the piston, obtained by the formula given on page 233.

If to the mean velocity for any fifth degree, as given in the

TABLE XVI.

Giving the rotative effect of a unit of pressure on the crank, at the termination of each interval of 5° in its entire revolution, counting either way from zero on the line of centres, when exerted through connecting-rods, the length of which is SIX *times,* FIVE *times, and* FOUR *times that of the crank.*

Degrees.	Connecting-rod = 6 cranks.			Connecting-rod = 5 cranks.			Connecting-rod = 4 cranks.		
	1. In the quadrants nearest to the cylinder.	2. In the quadrants farthest from the cylinder.	Difference.	1. In the quadrants nearest to the cylinder.	2. In the quadrants farthest from the cylinder.	Difference.	1. In the quadrants nearest to the cylinder.	2. In the quadrants farthest from the cylinder.	Difference.
5°	·10162	·07269	·02893	·10452	·06979	·03473	·10887	·06545	·04342
10	·20216	·14514	·95702	·20788	·13942	·06846	·21643	·13087	·08556
15	·30053	·21713	·08340	·30888	·20877	·10011	·32146	·19620	·12526
20	·39567	·28838	·10729	·40645	·27761	·12884	·42266	·26138	·16128
25	·48662	·35863	·12799	·49950	·34576	·15374	·51890	·32636	·19254
30	·57241	·42760	·14481	·58704	·41297	·17407	·60910	·39091	·21819
35	·65225	·49492	·15733	·66816	·47901	·18915	·69226	·45492	·23734
40	·72533	·56028	·16505	·74210	·54351	·19859	·76750	·51810	·24940
45	·79103	·62321	·16782	·80812	·60612	·20200	·83411	·58014	·25397
50	·84878	·68334	·16544	·86571	·66642	·19929	·89146	·64065	·25081
55	·89822	·74013	·15809	·91442	·72392	·19050	·93918	·69918	·24000
60	·93896	·79312	·14584	·95397	·77812	·17585	·97691	·75517	·22174
65	·97089	·84175	·12914	·98421	·82845	·15576	1·00462	·80803	·19659
70	·99394	·88549	·10845	1·00515	·87429	·13086	1·02236	·85707	·16529
75	1·00816	·92373	·08443	1·01690	·91500	·10190	1·03035	·90155	·12880
80	1·01372	·95594	·05778	1·01989	·94995	·06994	1·02894	·94073	·08821
85	1·01089	·98155	·02934	1·01392	·97850	·03542	1·01863	·97381	·04482
90°	1·00000	1·00000		1·00000	1·00000		1·00000	1·00000	

second column of the Tables at the end of the book, there be added one-half the acceleration for that degree, and the terminal velocity thus found be multiplied by the constant 57·3, with which we are familiar—expressing the ratio which unity bears to the velocity of the crank per degree of arc, or to the velocity of the piston when the crank is at 90°—the product will be the coefficient given in this Table for the same proportionate length of connecting-rod; the rotative effect of the pressure maintaining a constant ratio with the velocity of the piston. In those Tables the rotative effect may be found, by the above method, for every degree.

On diagram No. 49, page 26, if the base-line E B be taken to represent the pressure on the crank at the end of any degree, then the length of the horizontal line representing the velocity of the piston at such point will also exhibit the rotative effect of such pressure.

2nd. The difference between the rotative effects of the same pressure at corresponding points in the opposite quadrants is 0 at the line of centre and at 90°, and culminates at the angle of 45°, where it reaches the following considerable amounts; namely, when the connecting-rod equals 6 cranks ·16782, when it equals 5 cranks ·202, and when it equals 4 cranks ·25379 of the entire pressure. If we compare the rotative effects at this point of the opposite quadrants with each other, the difference will be found to amount, in the above three cases respectively, to 26·9 per cent., 33·3 per cent., and 43·8 per cent. This difference is the same at all speeds of the piston.

3rd. At the angle of 90°, at which point the piston and the extremity of the crank are moving with the same velocity, the rotative effect is equal to the pressure; and for a considerable distance from this point, in the quadrants nearest to the cylinder, it is greater than the pressure as the velocity of the piston is then greater than that of the crank.

The angular vibration of the connecting-rod produces two entirely separate effects, the distinction between which should be clearly apprehended.

First, as shown in the preceding section, it increases the acceleration and retardation of the piston in the forward end of the cylinder and diminishes them in the back end, and thus modifies the distribution through the stroke of the pressure exerted on the crank; and this effect varies in amount, as there explained, according to the weight of the transmitting parts,

and the speed and the length of the stroke, as well as with the proportionate length of the rod.

Second, as shown in this section, it causes the rotative effect of the pressure on the crank to differ in the opposite quadrants; and this effect is *not* varied by the speed or length of stroke, or weight of the transmitting parts, but only by the proportionate length, and consequent degree of angular vibration of the rod.

SECTION V.

OF THE TRANSMITTING PARTS OF AN ENGINE, CONSIDERED AS AN EQUALISER OF MOTION.

The investigation through which we have passed in the preceding sections enables us to perceive the adaptation of these parts of an engine to an office widely remote from that which they are primarily, and which hitherto they have been solely, intended to perform; namely, that of an equaliser of motion.

The functions of a fly-wheel have been three: first, to overcome sudden resistances; second, to give the governor time to act; and third, to maintain approximately uniform motion during a revolution; compensating for the extreme differences in the rotative effect of the steam pressure at different points, by absorbing the excess when the crank is in its most favourable positions, and giving it out again when it is in its most unfavourable ones.

The second of these offices has been abolished. The best governors now in use require no time to act. Their action in varying the pressure of steam admitted to the cylinder is coincident with the changes of resistance, however sudden or extreme these may be.

On the other hand, the improvements in the manner of working steam, admitting it at a higher pressure, cutting off early, and expanding to a low point at the end of the stroke, have, to the varying rotative effect of a constant pressure, added enormous differences of pressure at the extremes of each stroke.

Thus the need of considerable inertia in the fly-wheel, increased as it is also by the equal liability of a sudden resistance to occur during the latter part of the stroke, has become greater than ever.

The power possessed by the transmitting parts of the engine of performing the functions of the fly-wheel is presented in the following Tables and diagrams. The degree in which the equalising action of these parts may enable size and weight in the fly-wheel to be dispensed with, and its special adaptation to the requirement, in this respect, of engines working steam most economically, as these are here exhibited, will certainly be regarded as remarkable.

The total steam pressure is taken at 102 lbs. on the square inch of piston, agreeing with that shown in diagrams 57 and 58 on pages 242 and 243. The rotative effects are obtained by multiplying the pressures, as figured on those diagrams, by the coefficients given in columns 1 and 2 of Table XVI. In the following diagrams these effects are measured on the radial lines and the scale is 80 lbs. to the inch, or one-half that employed in the above diagrams of pressure.

Table XVII.

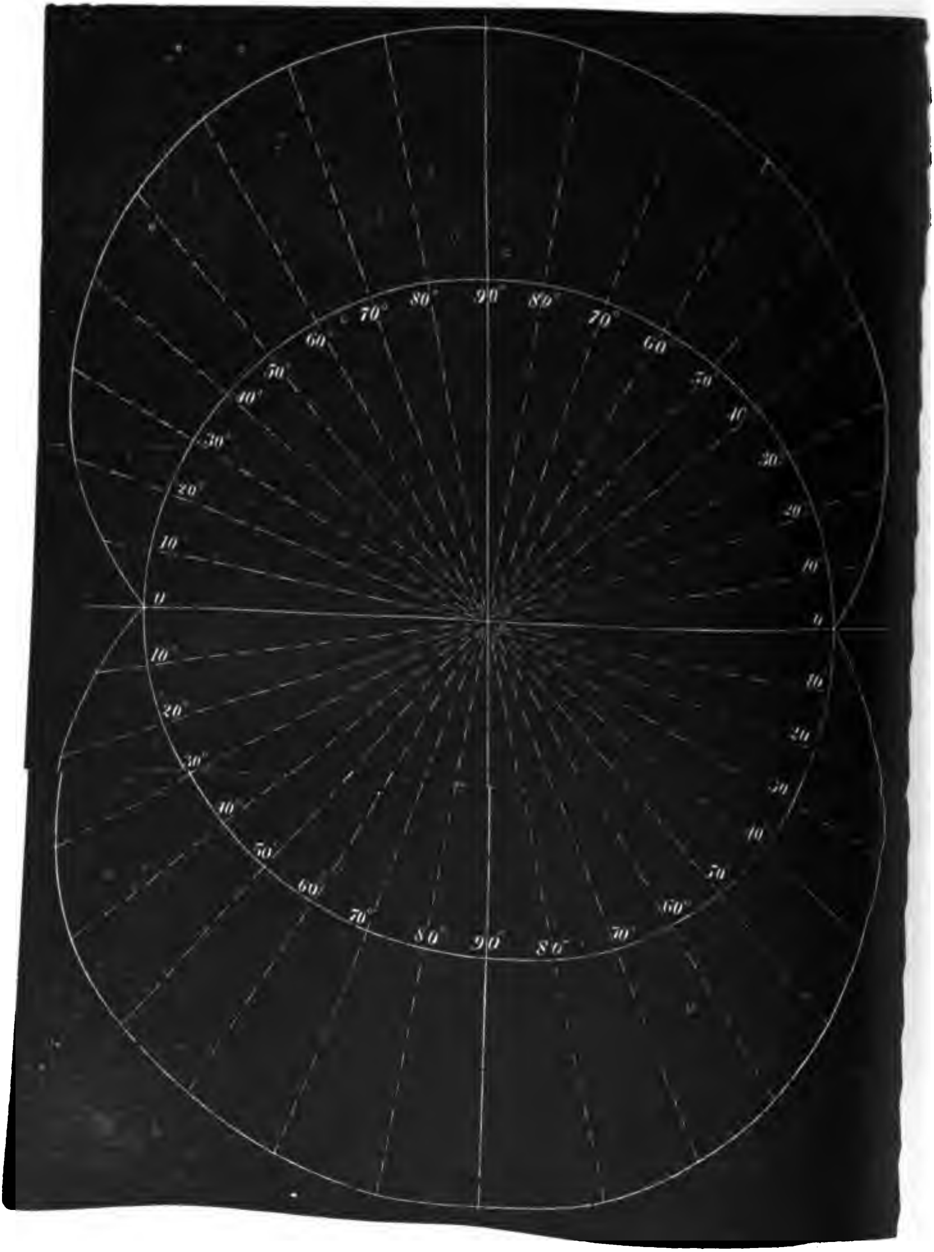
Giving the rotative effect of a pressure on the piston of 102 *lbs. on the square inch, in the following cases :—*

(*The forward stroke to be read* DOWNWARD, *and the return stroke* UPWARD.)

Degrees.	1st. Unaffected by the inertia of the transmitting parts	2nd.		3rd. Cut off at one-sixth of the stroke, and modified, as shown in the same diagrams, by the inertia of the transmitting parts.	
	Constant.	Cut off at one-sixth of the stroke, and forming diagrams Nos. 57 & 58.			
	Both Strokes.	Forward.	Return.	Forward.	Return.
	(*a*)	(*b*)	(*c*)	(*d*)	(*e*)
0°	0·0	0·0	0·0	0·0	0·0
10	14·79	14·21	·335	8·7	5·66
20	29·58	29·29	3·16	18·72	14·
30	43·66	43·66	7·1	28·68	22·5
40	57·12	57·12	10·64	38·36	29·4
45	63·55	63·55	12·46	43·76	32·4
50	69·66	69·36	14·0	48·5	35·2
60	80·88	61·62	17·77	40·44	38·86
70	90·27	53·1	22·1	33·63	41·14
80	97·5	46·37	27·36	31·55	42·06
90	102·	40·	33·5	32·	41 5
80	103·4	33·46	40·56	34·5	40·
70	101·4	28·33	49·7	38·27	39·76
60	95·88	23·5	59·22	41·32	41·3
50	86·7	19·10	71·4	42·87	48·4
45	80·68	17·20	79·	42·32	53·78
40	73·94	15·22	73·94	41·06	47·85
30	58·14	10·87	58·14	35·5	33·46
20	40·8	5·6	40·8	24·95	21·
10	20·6	·81	20·0	10·10	9·5
0°	0·0	0·0	0·0	0·0	0·0

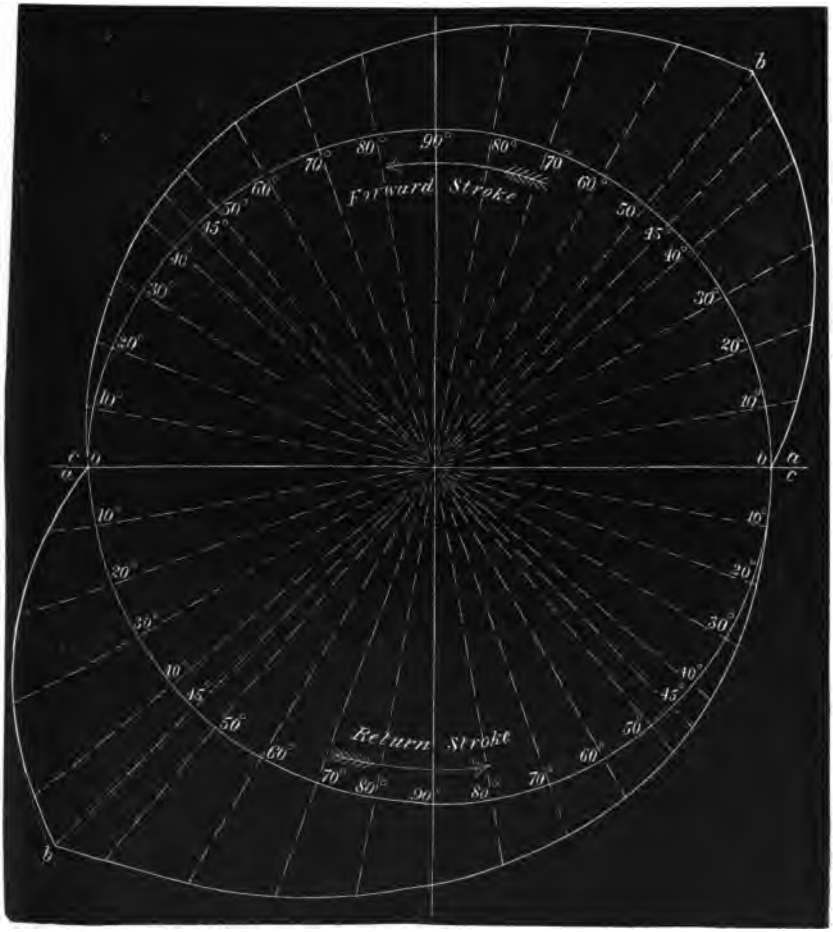
Column (*a*) of this Table and diagram No. 61 represent the

No. 61.



rotative effect of this pressure, if exerted uniformly through the stroke, and communicated to the crank without sensible modification from the inertia of the transmitting parts.

No. 62.



Columns (*b*) and (*c*) and diagram No. 62 represent the rotative effect of the same pressure cut off at one-sixth of the stroke, and also conceived to be transmitted without modification to the

crank. Columns (*d*) and (*e*) and diagram No. 63 represent the same pressure cut off at the same point of the stroke, but modified in its transmission to the crank by the inertia of the transmitting parts, in the degree represented in diagrams 57 and 58.

These two last diagrams tell their own story. In the first one we see the single impulse in each half-revolution culminating at the opposite points *b b*, and in the second we find this changed into an approximation to uniformity of rotative effect, which we shall as we proceed see increasing reason to admire.

If we consider these diagrams we shall observe, that in the first one, if the steam is cut off at an earlier point than one-sixth of the stroke, then the line *a b* will be proportionately shortened, and to represent the rotative effect produced during the expansion, a line in place of the line *b c* will be drawn, nearly parallel with that line but nearer to the circle; so that the difference between the rotative effects in the two quadrants of the half-revolution will be still greater than is here shown, and will become greater and greater the earlier the steam is cut off: while in diagram No. 63 a similar shortening, or a lengthening in a considerable degree, of the line *a b*, and change of position of the line *b c*, by variation of the point of cut-off, will make comparatively little difference in the uniformity of rotative effect.

But the approximation to uniformity of rotative effect obtained in this manner cannot be estimated at its real value, unless it shall be compared with that which is ordinarily obtained by the use of coupled engines.

It is generally considered that in this respect coupled engines possess a twofold advantage: first, in that the aggregate rotative effect is nearly uniform throughout the revolution; and second, in that the variations which do occur alternate four times during a revolution: so that in engines working expansively the rotative effect in the two quadrants of each half-revolution, instead of being extremely different, is the same, or nearly so.

The last is the great point of advantage of coupled engines over single-cylinder engines making the same number of revolutions per minute, and which enables them to be run with light fly-wheels without sensible variation in the motion.

With respect to the first point, however, we shall see that

the approach to uniformity of rotative effect attained by the use of coupled engines is much less than is commonly supposed.

No. 63.

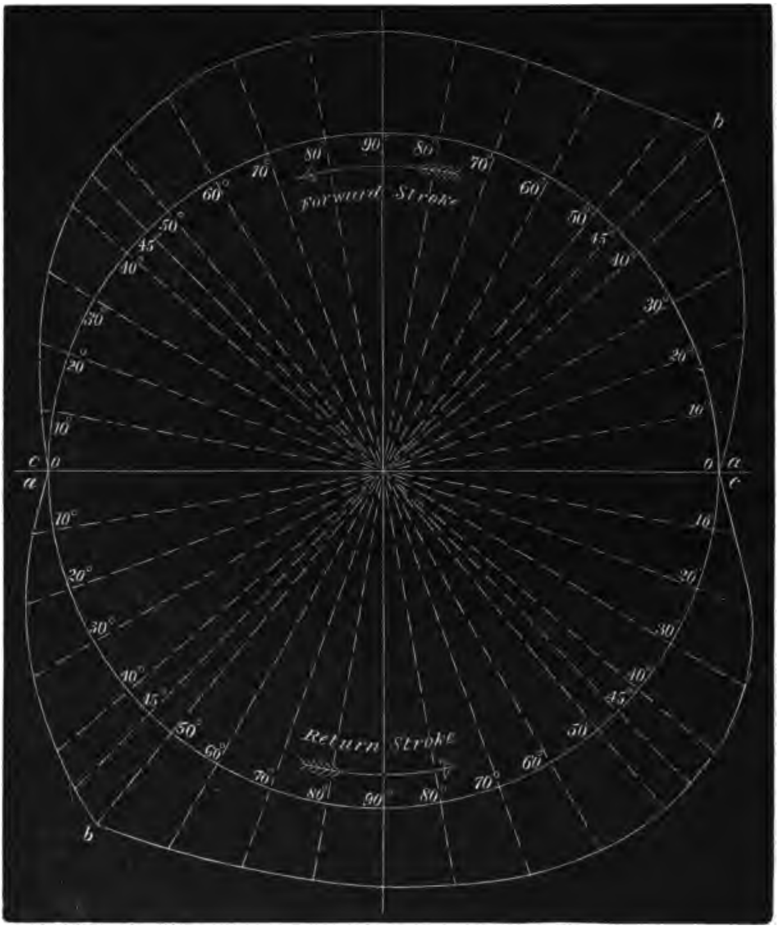
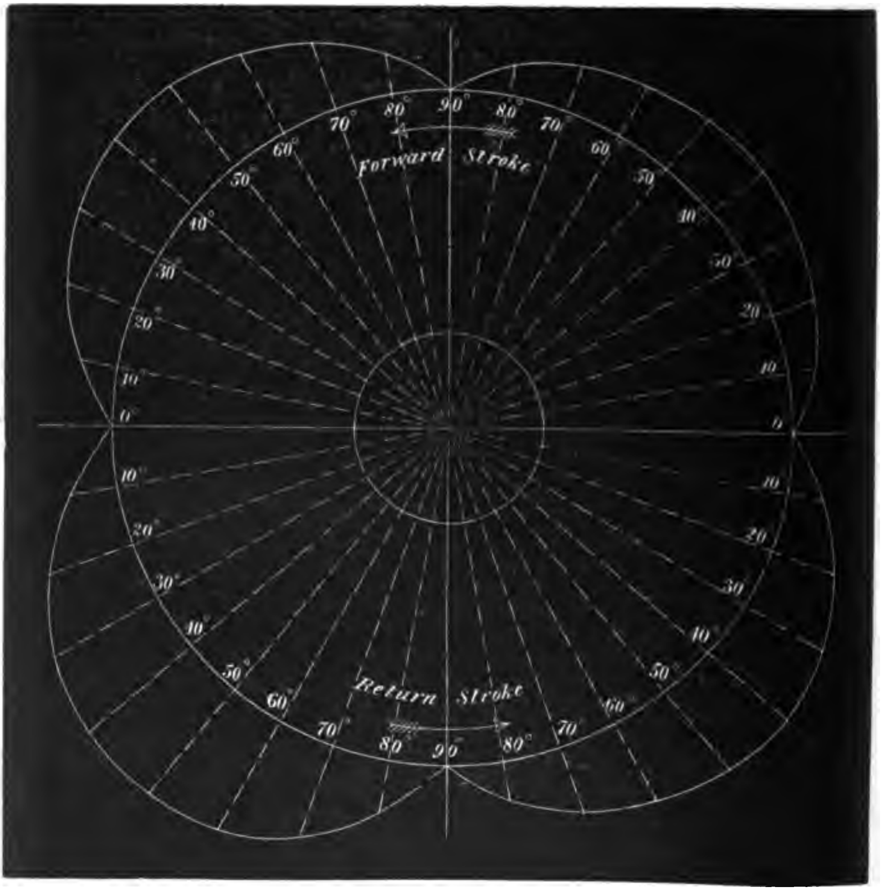


Diagram No. 64 exhibits the variations of rotative effect, in the case of coupled engines in which the steam follows the piston to the end of the stroke, as found by adding together two diagrams similar to No. 61, when set at right angles with each other.

The total rotative effects are measured from the inner circle; the excessive amounts are shown beyond the outer one on the same scale of 80 lbs. to the inch, which is the scale used through-

No. 64.

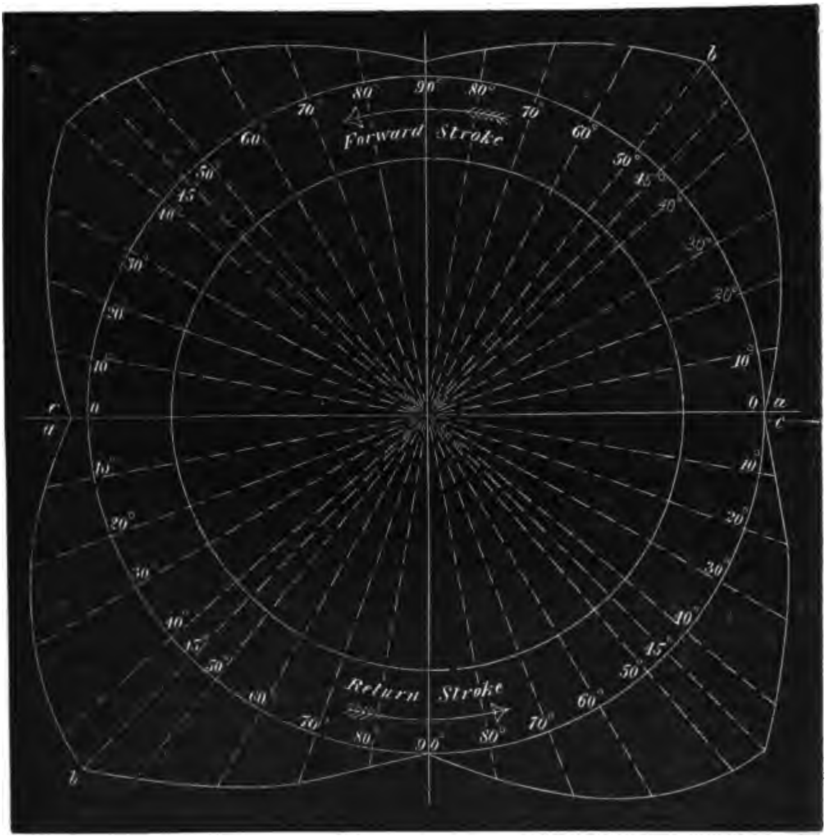


out this section. It is these variations only, which the fly-wheel is employed to compensate, with which we are here concerned; and we observe that the excess of the highest over the lowest rotative effect is equal to 56 lbs. on the square inch of

piston, or some 2 lbs. more than the highest pressure shown on diagram No. 63.

Diagram No. 65 exhibits the variations in the rotative effect

No. 65.



produced by coupled engines, each making the diagrams of steam-pressure shown in No. 47, and that of rotative effect shown in No. 62. The total effects are measured from the inner circle. These variations, it will be observed, are incomparably more serious than those shown in diagram No. 63.

It has been deemed desirable to exhibit an accurate compari-

son of these two diagrams, Nos. 63 and 65, and Table XVIII. has been prepared for this purpose.

Line A, in this Table, shows the rotative effect of the pressure on crank No. 1 of the coupled engines, at the termination of each interval of 10° in its revolution, and corresponds to diagram No. 62. Line B shows the rotative effects on crank No. 2 at the same instants of time. Line C is the sum of lines A and B, and shows the aggregate rotative effects at these points in the revolution of the shaft. Line D (constant) is the least of these aggregate effects, which occurs twice in the revolution, and which being subtracted from the line C, the difference is the excess above this least amount at each other point. These are seen in line E, and are shown, beyond the outer circle, in diagram No. 65. Below these, in line F, the corresponding rotative effects, shown in diagram No. 63, are given, copied from columns (*d*) and (*e*) of Table No. XVII., for ready comparison. The difference in uniformity in favour of the latter will be seen to be surprising.

The fact appears, that if in these two cases the engines could be run to make the same number of revolutions per minute, then the enormous disadvantage in point of uniformity of rotative effect which diagram No. 62 shows in comparison with diagram No. 63, would be remedied by coupling two engines, only in the degree represented in diagram No. 65.

It is curious to observe, that if engines are coupled at right angles, the inequalities in the rotative effects are nearly the same, whether the inertia of the transmitting parts be little or great in amount. This may be illustrated by showing the diagram of rotative effect got by coupling two engines, each producing diagrams similar to No. 63. Two such diagrams, set at 90° with each other, and added together, give the aggregate effects shown in the above diagram, No. 66. The total effects are measured from the inner circle. The explanation of the distortion is quite simple. The rotative effects on the two cranks at 45° coincide, and those on one crank at 0° coincide with those on the other at 90°.

Two things are here made obvious: First, it would be a very foolish proceeding to couple two of these engines with the idea of improving the uniformity of rotation; and, second, if this were done, the result would be quite as good as that which would be got by coupling the same engines, if the inertia could be got rid of, and each one made diagram No. 62; for the two

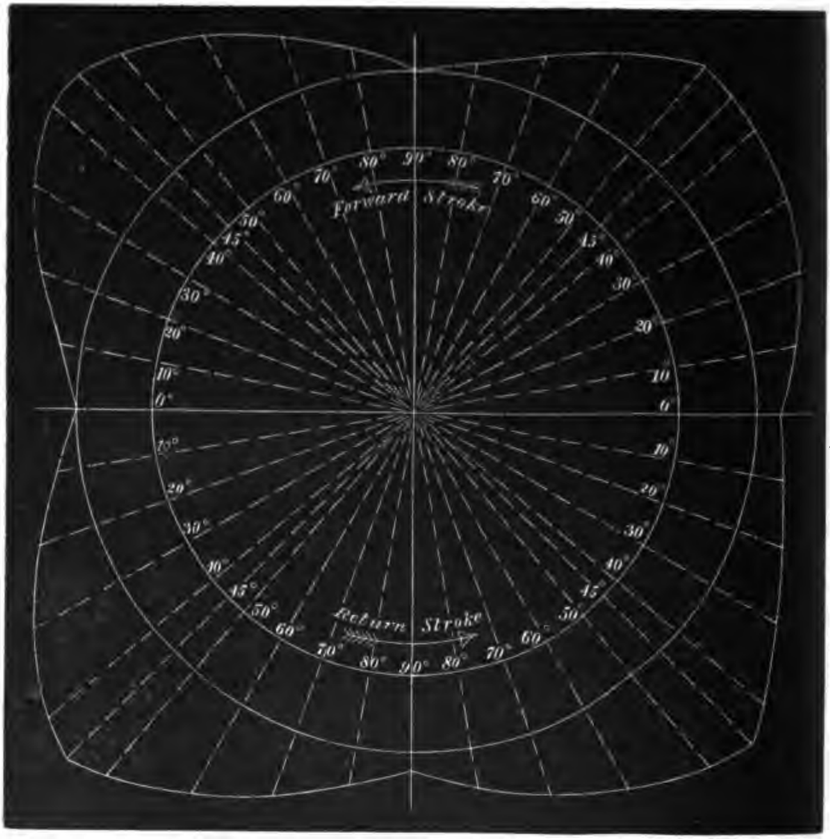
TABLE XVIII.

Exhibiting the amount of variation from a uniform rotative effect, when coupled engines are employed, with cranks set at the angle of 90° with each other, and each making diagram No. 47, and conceived to be unaffected by the inertia of the transmitting parts; in comparison with this variation when a single engine is employed, making the same diagram, but with the pressure on the crank modified, as represented in diagrams Nos. 57 and 58, by the inertia of these parts of the engine.

	0°	10°	20°	30°	40°	45°	50°	60°	70°	80°	90°	80°	70°	60°	50°	45°	40°	30°	20°	10°	0°
A	0·	14·21	29·29	43·66	57·12	63·55	69·36	61·62	53·1	46·37	40·	33·46	28·33	23·5	19·10	17·20	15·22	10·87	5·6	·81	
B	33·5	27·36	22·1	17·77	14·	12·46	10·64	7·1	3·16	·33	0·	14·21	29·29	43·66	57·12	63·55	69·36	61·62	53·1	46·37	
C	33·5	41·57	51·39	61·43	71·12	76·01	80·	68·72	56·26	46·70	40·	47·67	57·62	67·16	76·22	80·75	84·58	72·49	58·7	47·18	
D	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	
E	0·	8·07	17·89	27·93	37·62	42·51	46·5	35·22	22·76	13·20	6·5	14·17	24·12	33·66	42·72	47·25	51·08	38·99	25·2	13·68	
F	0·	8·7	18·72	28·68	38·36	43·76	48·5	40·44	33·63	31·55	32·	34·5	38·27	41·32	42·87	42·32	41·06	35·5	24·95	10·10	
A		·33	3·16	7·1	10·64	12·46	14·	17·77	22·1	27·36	33·5	40·56	49·7	59·22	71·4	79·	73·94	58·14	40·8	20·	0·
B		40·56	49·7	59·22	71·4	79·	73·94	58·14	40·8	20·	0·	·81	5·6	10·87	15·22	17·20	19·10	23·5	28·33	33·46	40·
C		40·89	52·86	66·32	82·04	91·46	87·94	75·91	62·9	47·36	33·5	41·37	55·3	70·09	86·62	96·20	93·04	81.64	69·13	53·46	40·
D		33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5	33·5
E		7·39	19·36	32·82	48·54	57·96	54·44	42·41	29·4	13·86	0·	7·87	21·8	36·59	53·12	62·70	59·54	48·14	35·63	19·96	6·5
F		5·63	14·	22·5	29·4	32·4	35·2	38·86	41·14	42·06	41·5	40·	39·76	41·3	48·4	53·78	47·85	33·46	21·	9·5	0·

diagrams, Nos. 65 and 66, are very much alike, showing the uniformity of rotative effect seen in diagram No. 63 to be impaired about as much as the want of it shown in diagram No. 62 is mended, by coupling the engines. It may be thought

No. 66.



that a better result can be got by setting the cranks at some other angle with each other than 90°. Trials have shown, however, that no other angle gives results so good as the above.

Surprising as are the results of the foregoing comparison, we

have as yet only begun to see the benefit which this action confers. The same rate of rotation has been assumed in the cases compared. In fact, however, the rates are very different. The amount of inertia in the transmitting parts required to produce the effects shown in diagram No. 63 is most readily developed by employing high-piston speed combined with a rather short stroke, and so giving rapid rotation to the shaft. The revolutions made by the engine here employed to illustrate this action are 140 per minute. (*See* pages 205–6 and 228.) Engines in ordinary use hitherto, to exert the same power, make from 50 to 60 revolutions per minute, but for convenience of comparison we will assume them to make 70 revolutions. At 140 per minute, one revolution is performed in the same time with a half-revolution at 70 per minute; and in comparing the irregularities in rotative effect in different cases, which must be compensated in order that uniform motion may be maintained, those which occur in equal intervals of time must be compared with each other.

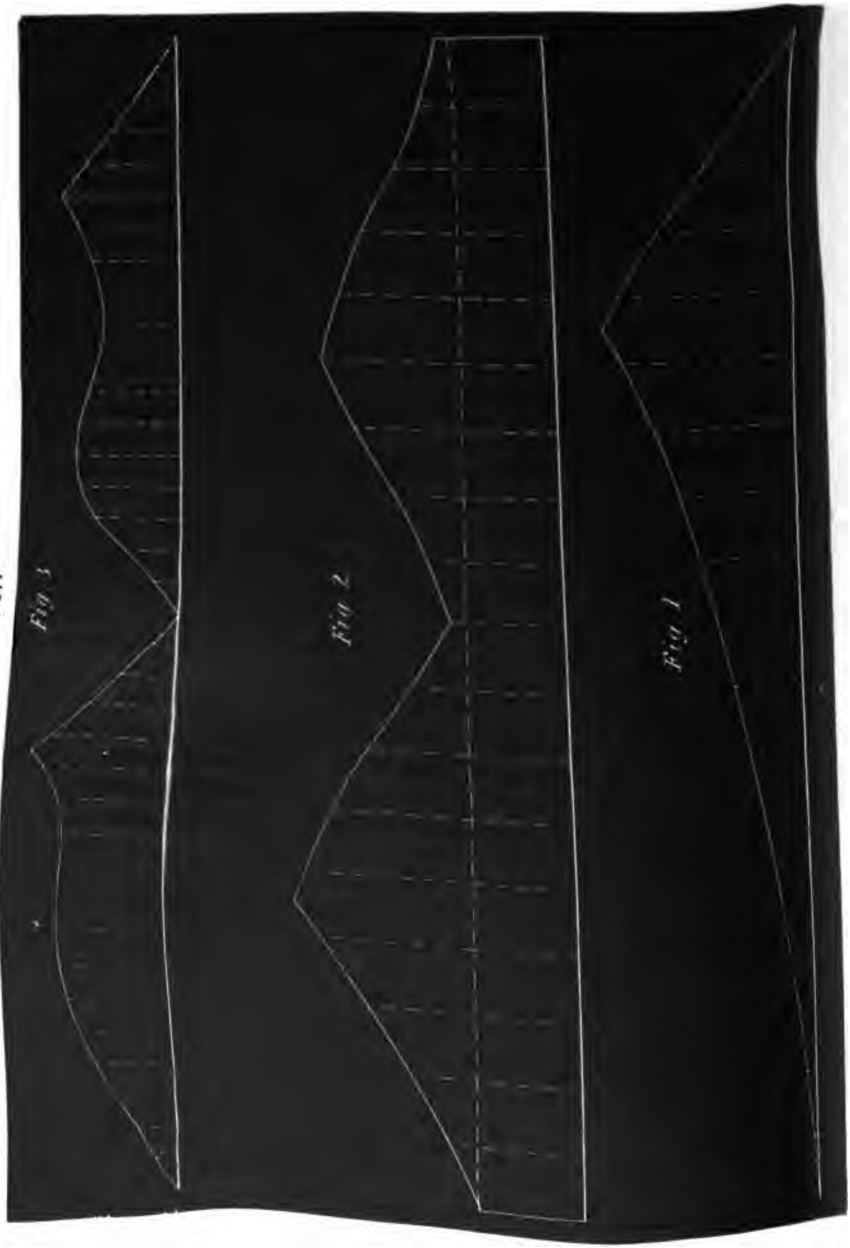
In diagram No. 67, on the next page, fig. 1 represents, during a half-revolution of the crank, the variation in the rotative effect resulting from cutting off at one-sixth of the stroke, in a single cylinder, the inertia of the transmitting parts being conceived to be insensible in amount. It corresponds to one-half of diagram No. 62.

Fig. 2 represents, in a similar manner, the variations in the total rotative effect during a half-revolution, when two engines, each producing the effect shown in fig. 1, are coupled at right angles. It corresponds to one-half of diagram No. 65. The aggregate effect is shown; the variations are measured from the dotted line.

Fig. 3 represents these variations during a complete revolution, performed in the same time with the half-revolution in the above cases, of a single engine, cutting off also at one-sixth of the stroke, but in which the inertia of the transmitting parts is made available in equalising the rotative effects. It corresponds to diagram No. 63. In all the figures dotted ordinates are drawn at intervals of 10°.

In comparing the inequalities in the last two cases, we see that those shown in fig. 3 have in reality only one-half the relative duration that is assumed in line F of Table XVIII. The rise and fall are rapid, and the approximately uniform effect continues during two-thirds of each semi-revolution;

No. 67.



while in fig. 2 we see excesses considerably greater separated by intervals twice as long.

The columns below exhibit together the real irregularities which occur in the two cases in equal times. Column (1) gives the rotative effects shown in fig. 3 at intervals of 10°. These, being for an entire revolution, continually repeat themselves; but the irregularities shown in diagram No. 65 are much greater during the return, than during the

	(1.)	(2.)	(3.)		(1.)	(2.)	(3.)
0	0·	0·	6·5				
10°	8·7			10°	5·66		
20	18·72	8·07	19·96	20	14·	13·68	7·39
30	28·68			30	22·5		
40	38·36	17·89	35·63	40	29·4	25·2	19·36
45	43·76			45	32·4		
50	48·5			50	35·2		
60	40·44	27·93	48·14	60	38·86	38·99	32·82
70	33·63			70	41·14		
80	31·55	37·62	59·54	80	42·06	51·08	48·54
90	32·	42·51	62·70	90	41·5	47·25	57·96
80	34·5	46·5	53·12	80	40·	42·72	54·44
70	38·27			70	39·76		
60	41·32	35·22	36·59	60	41·3	33·66	42·41
50	42·87			50	48·4		
45	42·32			45	53·78		
40	41·06	22·76	21·8	40	47·85	24·12	29·4
30	35·5			30	33·46		
20	24·95	13·20	7·87	20	21·	14·17	13·86
10°	10·10			10°	9·5		
				0	0·	6·5	·0

forward, stroke of the piston. In fig. 2, diagram 67, these irregularities for the first and third quadrants, being respectively the least and the greatest, are shown. In this tabular comparison, however, all should be given; therefore column (2) gives the excesses of rotative effect as shown on diagram No. 65 during the forward, and column (3) gives those during the return, stroke.

A careful study of this Table and the figures will give a

correct apprehension of the great difference in the amount of regulating power required to produce an equally close approximation to uniform rotation in the two cases.

But, moreover, in the case of the single engine the fly-wheel makes two revolutions, while in that of the coupled engine it is making one. The dynamical energy, or living force, of the same wheel is therefore four times as great in the former case as it is in the latter; or, with the same diameter, one-fourth of the weight possesses the same equalising power. If, however, instead of reducing the weight, the diameter of the wheel were in the former case reduced one-half, and its weight kept the same, we should then, in the two cases, have equal weights moving with the same velocity, and so possessing the same equalising power.

The increased speed, then, enables us also to employ, to exert a given equalising effect, a fly-wheel of only one-fourth the weight with the same diameter, or of one-half the diameter with the same weight—the equalising power at a given number of revolutions per minute, of a wheel of a given section of rim varying as the cube of the diameter.

This comparison, it must not be forgotten, is made with the light fly-wheel required by coupled engines making 70 revolutions per minute. To estimate correctly, however, the extent to which this development of the equalising power of the transmitting parts enables the fly-wheel to be dispensed with, the comparison must be made with the requirement of a single-cylinder engine, making about 50 revolutions per minute, and of a size sufficient to give, at that speed, the power that is furnished by the coupled engines at 70 revolutions, or by a single engine of the same dimensions at 140 revolutions per minute.

These results are obtained mainly by the employment of rapid reciprocation—frequent changes in the direction of the piston's motion—an action which has been the traditional dread of engineers, but a proper understanding of which strips it of its terrors, and will hereafter enable it to be employed with this excellent advantage.

A RIDE ON A BUFFER-BEAM.

THE following account of a ride on a buffer-beam was printed as an Appendix to the first edition:—

While this pamphlet was in the press, the Author enjoyed the privilege, through the courtesy of Mr. Sinclair, Engineer of the Great Eastern Railway, of making a trip from London to Yarmouth and back, in company with Mr. Zerah Colburn, on the buffer-beam of an express engine; and it has occurred to him that an account of the method employed for taking diagrams and making the necessary observations would probably be interesting and useful to those who might wish to apply the Indicator in a similar manner.

The locomotive was one of the largest class, having outside cylinders of 16 in. bore and 24 in. stroke, and 7 ft. 1 in. driving-wheels, making 237 revolutions to the mile. The number of revolutions actually made rose occasionally to 250, and even to 260 per minute, the latter giving a speed at the rate of 66 miles per hour. The diagrams were taken from the forward end only of each cylinder. Short bent pipes, of $\frac{5}{8}''$ internal diameter, were screwed into the cylinder-covers as near to the upper side as possible, and to the ends of these the Indicators were attached. A platform was laid over the buffer-beam, and enclosed with a stout iron railing, the ends of which were bolted to the smoke-box. Comfortable seats were provided, on one of which each operator sat, quite secure, with his back to the wind and the Indicator between his knees.

The method employed for giving motion to the paper was very simple. A bar of light angle-iron, bent in the form Λ, was bolted on the upper guide-bar; and at the apex, about three feet above the centre line of the engine, a pin was set, projecting horizontally outward. A light-arm swung from this pin, and received a vibratory motion from a pin projecting from the cross-head, and working in a slot in the lower end of the arm. A button-headed pin was inserted in this arm, about

7″ below the point of suspension, and to this a cord was attached, leading directly to the Indicator, giving to the paper a motion of about 4½″. It had been found, at the highest speeds, to be a very troublesome operation to hook on to the loop at the end of this cord with the small hook with which the Indicator cord is furnished; a brass ring an inch and a quarter in diameter was therefore substituted for the loop, and a hook provided of corresponding size. The ring was also secured in position by two other cords, so that it could move the required distance back and forth, but would not fall, when disengaged, where it could not be readily seized. By this means it was found that the connection could easily be made at any speed.

It was arranged with the engine-driver that he should not run at any time with the throttle-valve partially closed, but should keep this quite open, when it was open at all, and should vary the power exerted by the engine, as should be required, entirely by changing the point of cut-off.

It was desirable that as large a number of diagrams should be taken as possible, because the number of carriages was changed at almost every station, the gradient up or down which the train was moving continually varied, curves of longer or shorter radius occurred occasionally, and the speed attained, and the point at which the steam was cut off, scarcely ever remained the same for more than a few minutes consecutively; so that, however frequently the diagrams might be taken, very few of them would be quite the same. The more nearly level and direct a road is, the less, of course, the changes in these conditions will be, both in amount and frequency. At the same time it was important to know precisely what the conditions were which would determine or affect the form of the diagram—namely, the pressure of steam, the point of cut-off, the number of revolutions making per minute, the gradient, the curve, and the weight of the train—at the instant when each diagram was taken.

The plan hit upon for obtaining these data was as follows:—The two operators applied themselves without interruption to taking diagrams, each being able to take about one per minute when moving slowly, and one in a minute and a half when going at full speed. Those from one cylinder were numbered in order, and those from the other were distinguished by letters, each repetition of the alphabet being numbered, commencing *a*, *a* 1, *a* 2, and so on. We were accompanied by

Mr. Maw, from Mr. Sinclair's office, who, standing between us, noted, in a book prepared for the purpose, the moment of passing each quarter-mile post, and of passing on to each gradient, and of taking each diagram, the number or letter being called out to him.

An assistant on the foot-plate of the engine noted, in like manner, the changes in the pressure of the steam, and of the notch of the quadrant in which the engine was run. The fuel consumed was likewise weighed, and the water measured, and at each station the number of carriages was noted. It became thus a very simple matter to minute afterwards on each diagram accurately all the conditions under which it was taken, and then to observe the power required to move different numbers of carriages at different speeds, and on different gradients, or around different curves; and at least one diagram being obtained on nearly every gradient and curve, and the time occupied in running over each being known, the whole power exerted during the trip could be very closely approximated to, and the consumption of fuel and of water per horse-power per hour could be ascertained, and of course the action of the steam at every speed and at each point of cut-off was completely revealed. Also the diagrams taken at an accelerating speed, when the momentum of the train was being accumulated, could be distinguished with certainty from those taken at a uniform speed, and which showed only the power required to maintain an equilibrium with the retarding forces. A remarkable feature of these diagrams was the very trifling back pressure exhibited, which was accounted for by the width of the ports 1½", and the size of the blast orifice, 5' diameter.

Diagrams from locomotive engines, on account of the great variety of speeds and points of cut-off at which they are taken, and the variations which they exhibit in the power exerted, are of higher general interest, in some respects, than those obtained from either stationary or marine engines; and a careful study of them may confidently be expected to throw light on some questions, about which engineers now differ in opinion. They show at once, for example, at what speed of piston a certain area of port ceases to be sufficient for a given diameter of cylinder, and precisely how velocity of piston in different degrees affects the pressure obtained.

This is illustrated in a remarkable manner in cut No. 19, where are shown two diagrams, taken on this trip, when the

engine was carrying the same pressure of steam, and was running in the same notch of the quadrant, and of course, therefore, cutting off the steam at the same point of the stroke. The diagram shown in the heavier line was taken at a speed

not exceeding 50 revolutions per minute, and the one distinguished by a lighter line was taken with the same instrument five minutes later, at the extreme velocity of 260 revolutions, or 1040 feet travel of piston per minute. The scale of

the Indicator was 40 lbs. to the inch, and the boiler pressure 120 lbs. on the square inch, which the more excessive compression made at the higher velocity caused for an instant to be nearly reached in the cylinder.

Much, among other things, may also be learned from these diagrams from locomotives upon that most important and vexed question—In what degree the cylinder acts as a condenser of the entering steam, and by what means and in what degree in non-condensing engines this vicious action may be corrected; and what, on the other hand, tends to aggravate it?

These diagrams are taken under fewer difficulties than would be at first imagined, if the weather is pleasant, and the proper provision is made for the comfort and security of the operators. The principal difficulty is from the wind, which, at very high speed, approaches more nearly to a hurricane than anything that one is able to experience in this latitude in any other way, and the labour of resisting it becomes quite wearisome, if the operator is not somewhat protected from its force. No unpleasant sensation whatever is produced by the rapid motion, the passing of trains is scarcely observed, and if no accident happens, there is no danger more than in the carriages. Good weather is essential to the satisfactory accomplishment of the objects of such an excursion.

THE "WAYNE"

STEAM ENGINE INDICATOR.

THIS Indicator has been designed to meet the long-felt want of a reliable instrument, to rapidly and accurately obtain diagrams from either slow or quick running engines.

Its advantages are:—

Small movement of parts connected to tracing point.

The piston and tracer fixed together as one solid piece.

No loose joints or parallel motion.

The bearings are free from friction.

The tracing pointer easily adjusted.

The spring being outside the cylinder, not subject to steam temperature.

The spring easily changed without moving any other part of the instrument.

The diagram paper very easily placed in position and removed more quickly than with any other indicator.

The control spring of the paper slide adjustable from the outside.

The moving parts very light.

With lining attachment, friction and inertia reduced to a minimum.

CONSTRUCTION.

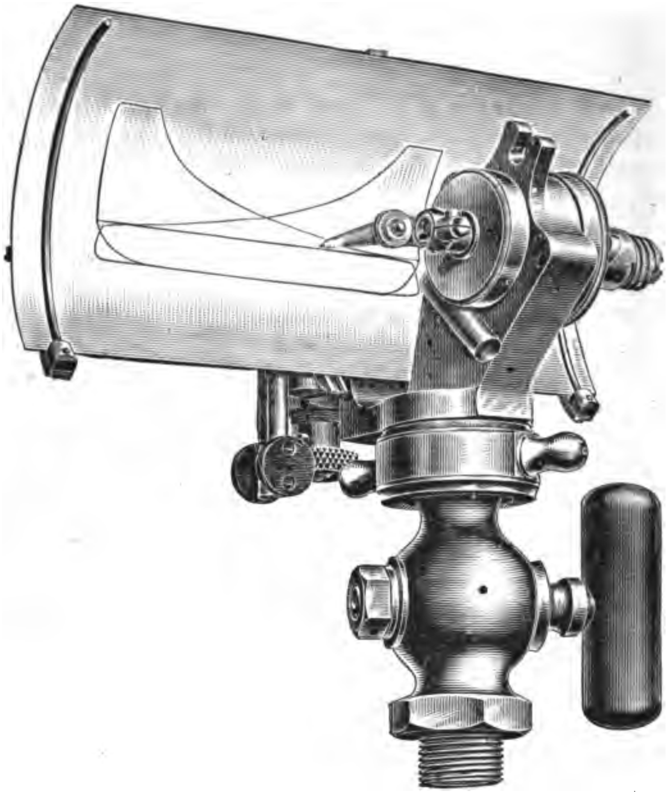
THE PISTON.

The piston consists of two tongues, or leaves, standing out from opposite sides of the piston rod, the rod passing through both ends of the cylinder; the leaves forming the piston reciprocate in a rotary manner between abutments from opposite sides of the inner periphery of the cylinder. These abutments fit the piston rod between the leaves of the piston; the steam, after passing the stopcock (which is connected to the indicator by ball fittings, thus allowing the indicator to be set at an angle), is divided, and each half enters the

cylinder near the abutments on opposite *sides* or diameters, thus acting upon both leaves of the piston with equal pressure; and the piston and piston rod, carrying the tracing point fixed on one end, are turned in a rotary direction.

With the reciprocating rotary motion there is very little

INDICATOR ARRANGED FOR TAKING PLAIN DIAGRAMS.



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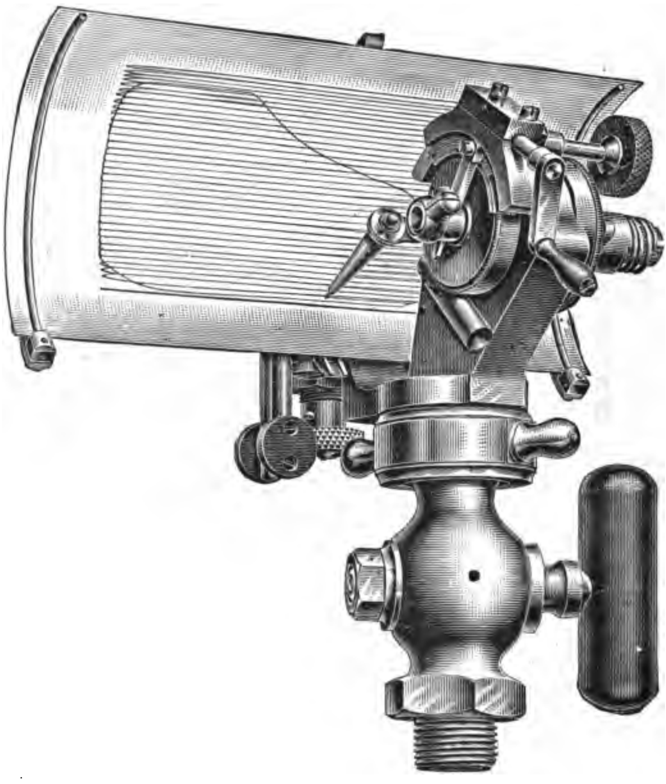
movement against gravity, as the piston and piston rod, carrying the tracer, only partly revolve in the fitting of the piston rod.

The piston is accurately turned, so that it is just free of the cylinder; it will therefore be seen that there is no sideway

pressure, and the piston cannot be driven against the walls of the cylinder by the escaping steam.

The piston, being steel, does not expand so much as the cylinder; there is, consequently, no fear of binding when hot.

INDICATOR ARRANGED FOR TAKING LINE DIAGRAMS.



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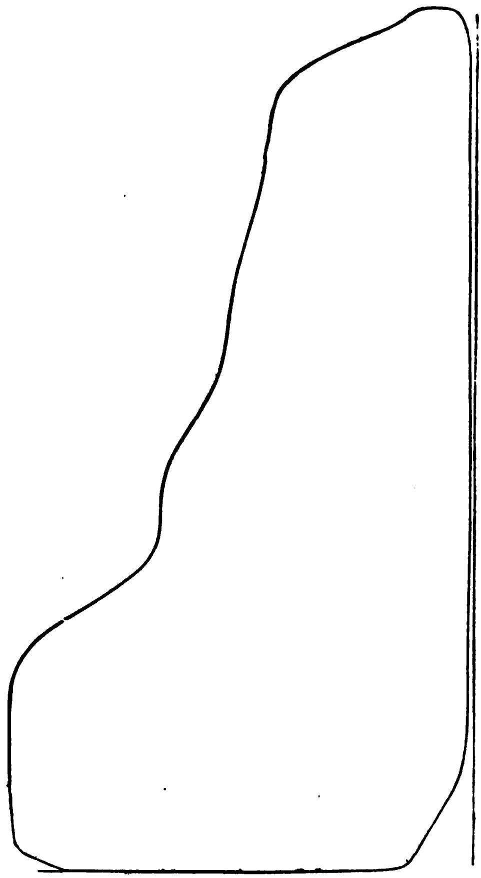
THE SPRING.

The spring is double coiled from one piece of wire; the cross piece of wire joining the two coils is held in a V slot in the end of the piston rod by a spiral grooved cap; the other ends of the spring are fixed to a small plate having two perforations, which fit on two steel pins standing out from one end

of the cylinder. The spring offering a resistance equally on both sides, the piston rod and tracer reciprocate in a rotary direction without friction.

Facsimile of Ordinary Diagram taken with the Wayne Indicator.



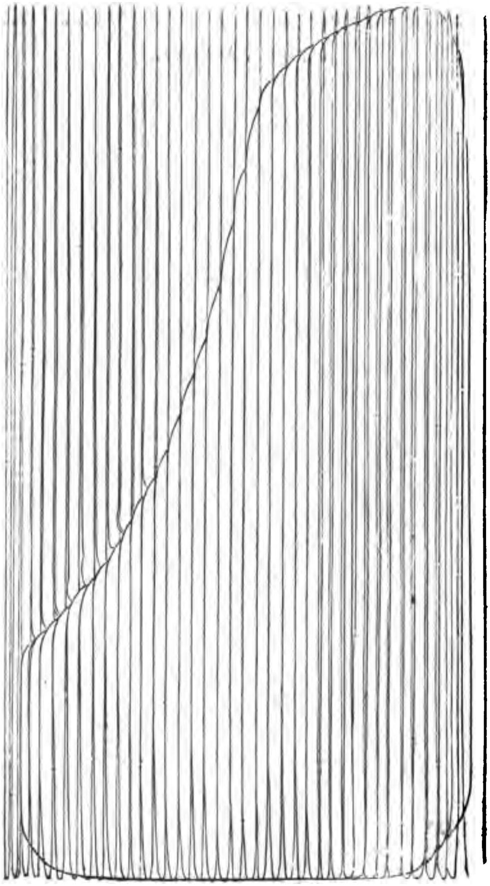
250 Revolutions per Minute.

The Tracer.

The tracer can have a brass point for metallic paper, lead point for plain paper, or a hard steel point for marking on our special black-faced paper; in the latter case there is no sharpening of points, and the diagram will not fade.

Paper Slide.

The paper is held in a cylindrical form, concentric with the piston rod, in light spring clips at each end of a sliding bar; from one end of the bar a cord is taken once round a pulley to which it is fixed. From the other end of the sliding bar a



Facsimile of Line Diagram taken with the Wayne Indicator.

250 Revolutions per Minute.

cord is taken round the other side of the pulley; then, passing between guide pulleys, it is attached to the engine in the usual way. The spindle on which the pulley revolves is a steel tube, the control spring passing through the centre, one end of the spring being fixed to the drum, the other end to a milled head,

having a cross pin to engage in recesses round the bottom edge of the steel tube, to regulate the strength of spring.

The drum and sliding bar are aluminium, and are very light.

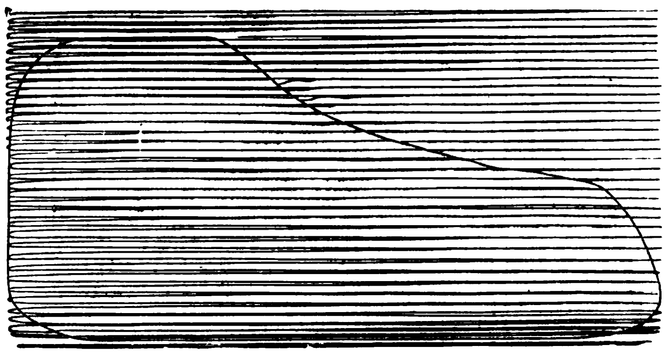
For very high speed (over 600 revolutions) the spring can

FACSIMILE OF DIAGRAMS TAKEN WITH THE WAYNE INDICATOR.

ORDINARY DIAGRAM.

500 *Revolutions per Minute.*

LINE DIAGRAM.



500 *Revolutions per Minute.*

be released and placed in a middle position, so that there may be a little torsion at each end of the stroke. A cord attached to each end of the sliding bar is passed round opposite sides of the drum, and connected to a disc or drum, which drum is reciprocated by a connection to the engine. By this arrangement very fine diagrams at high speeds can be taken.

LINE DIAGRAM ATTACHMENT.

An arrangement is provided for taking the diagram in parts or lines, with a mechanically limited stroke for increase of

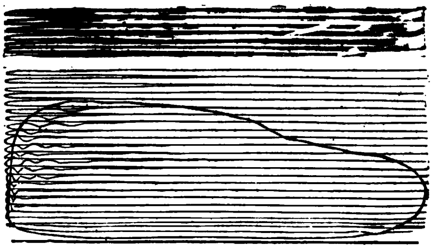
pressure at the steam end of the line, and a corresponding down stroke on the expansion part of the line. This limiting is arranged by a segment of a worm wheel mounted concentric with the piston rod.

A steel tongue, mounted on a worm wheel, passes through a hole in the piston rod, and is attached to the worm wheel by a screw passing through an elongated hole in the steel tongue, so that for each fall and rise of steam pressure, the piston can only rotate through an angle equal to the loss of time in the hole. The tracer moves about one-eighth of an inch for each

ORDINARY DIAGRAM.

1000 *Revolutions per Minute.*

LINE DIAGRAM.



1000 *Revolutions per Minute.*

stroke of the engine. By turning the worm from the atmospheric line, the tracer will move over the full range of the diagram.

This lining arrangement is put on or taken off in a few seconds; the stem is slipped between the horns provided on top of the cylinder, the pin passed through a hole in the piston rod, and the milled head screwed up.

The advantage of the lining or checking arrangement is, that the indicator will give a correct diagram at very high speeds; and at all speeds weaker springs may be used, giving large and accurate diagrams.

INSTRUCTIONS FOR WORKING.

Use only thin clean oil as a lubricant; we recommend gas engine cylinder oil.

In using the Indicator, first see that it is well cleaned and free from dirt. This can easily be proved at any time by taking off the spring and trying if the piston is free.

Well blow out the steam ways, and fix the Indicator on the cock in the most convenient position for use, so that the cord will have the straightest run to the engine attachment.

Adjust pressure of tracer on paper, which can be done by the screw nut at the bottom of cone; if the point is just clear of the paper before bringing the slide forward, it will be right: before putting in the paper, bring the paper slide to its forward position; then press the tracing point out to the bottom of the recess in the fixed curved guide. The point should be in the centre of this recess; if it comes on one edge it may tear the paper.

If necessary, adjust pressure with the screwed nut; this nut must not be loose.

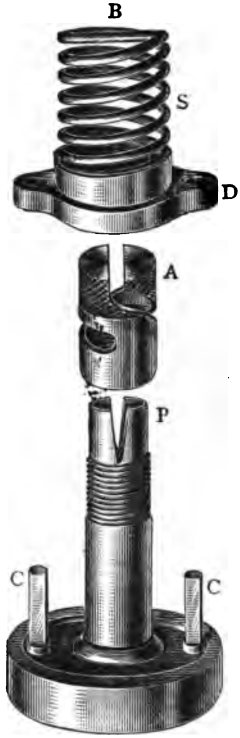
The centre wire passes through a small tube or sleeve; the end of this tube is sprung, and should clip the wire tight enough to prevent it slipping, and is intended for a coarse adjustment of the point. This tube plunges freely through the nut, and is held forward by the light spiral spring. The nut is screwed into the large end of the taper tube; by screwing this nut in or out gives a fine adjustment to the point. The wire must not be bent. Before taking a diagram, see that the wire and its tube fittings are free. The wire can be withdrawn for sharpening. All these parts must be quite clean, as dirt will prevent the plunging movement and tear the paper. The wire and nut may be taken out, and all the parts cleaned in a pot of paraffin.

To place the paper in position, take the paper by the lower corners with each hand, keeping the paper taut, and pass it down the spring clips and parallel with the slide; if working at high speeds, give a little rubbing pressure on the curved springs, which will press the paper in the recess, and make it impossible to slip.

When using the white paper, the brass point soon gets

polished, and increase of pressure will not make it mark; the point should be re-sharpened and the polished surface taken off.

The Indicator should be well cleaned and dried directly after use, as the steel piston may rust; if the steel wire has been used in the tracer this should also be cleaned and dried.



Full size.

Always use the hard steel point for black paper, which gives a very fine line.

When placing the spring on the instrument, first see that the cross slot of the cap A corresponds with the V slot in the piston rod P; then place the cross wire B, of the spring S, in the slot, and turn the piston rod to bring the holes in the spring plate D to the two steel studs C (the spring plate D should slide on these studs freely but without shake); then press the spring on, and turn the cap A on the piston rod; the spiral grooves will then press the cross wire of the spring in the V slot (the handle of the screw-driver is arranged for turning this cap).

When using the strong springs, see that the cap is not screwed too far on the piston rod P, or the cross wire will fit the V before the spiral grooves can engage the wire, and when fine springs are used, unless the cap is screwed on far enough, the cross wire may come to the end of the spiral grooves instead of being pressed down in the V; the tapped end of the cap A should be kept sprung in, so that it will fit the piston rod tightly but evenly. (*For very high speed we advise special springs.*)

Warm up the Indicator and take atmospheric line.

When ready, bring the paper slide to its forward position.

Care must be taken that the proper torsion is given on the drum spring for high speeds by turning the milled nut at the bottom of steel spindle. If the spring is left with too great a torsion, the diagram will be distorted by the leaning of the

steam line in one direction, and if too little torsion be given, by a leaning in the opposite direction.

As the instruments are sent out, the drum springs will be right for speeds up to 500.

The paper slide must be kept clean and quite free.

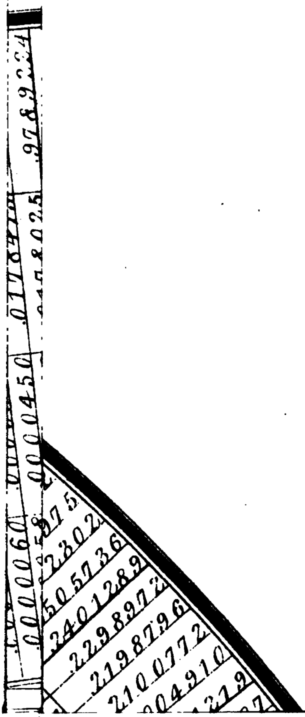
To take a diagram with the lining arrangement, first find the atmospheric line by turning the worm till the tracer can be moved a little up and down by pressing it; in other words, the elongated hole in the steel tongue must move freely on the screw fixed to the worm wheel.

The atmospheric line should then be taken (or the lining arrangement may be removed to take the atmospheric line). The steam should then be turned on, and the paper slide brought forward; the worm should next be turned by the small handle till the diagram is complete—that is, till the steam is not strong enough to lift the tracer. The best speed to turn the worm can easily be determined by experiment according to the speed of revolution of the engine; as a general rule it will be found convenient to make the diagram with about 18 to 20 lines per inch of height.

LIST OF SPRINGS FOR WAYNE'S INDICATOR.

English Measure.				Metric Measure.		
No. of Spring.	Scale per lb. per square inch.	Range in Pounds.		Scale per Kilo. per square c/m.	Range in Kilogrammes.	
		Vacuum.	Pressure.		Vacuum.	Pressure.
				Millimetres.		
1	$\frac{1}{6}$	−15	+ 0	60·21	−1	+ 0·05
2	$\frac{1}{8}$	−15	+ 5	45·16	−1	+ 0·41
3	$\frac{1}{12}$	−15	+ 15	30·10	−1	+ 1·11
4	$\frac{1}{16}$	−15	+ 25	22·58	−1	+ 1·81
5	$\frac{1}{24}$	−15	+ 45	15·05	−1	+ 3·21
6	$\frac{1}{36}$	−15	+ 90	10·03	−1	+ 6·33
7	$\frac{1}{46}$	−15	+115	7·85	−1	+ 8·09
8	$\frac{1}{56}$	−15	+140	6·45	−1	+ 9·84
9	$\frac{1}{66}$	−15	+165	5·47	−1	+11·60
10	$\frac{1}{76}$	−15	+190	4·75	−1	+13·37
11	$\frac{1}{86}$	−15	+215	4·20	−1	+15·12
12	$\frac{1}{96}$	−15	+240	3·76	−1	+16·89
A	$\frac{1}{5}$	−12·5	+ 0	72·25	−0·879	+ 0·00
B	$\frac{1}{10}$	−15	+ 10	36·12	−1	+ 0·76
C	$\frac{1}{15}$	−15	+ 22·5	24·08	−1	+ 1·64
D	$\frac{1}{20}$	−15	+ 35	18·06	−1	+ 2·52
E	$\frac{1}{25}$	−15	+ 47·5	14·45	−1	+ 3·39
F	$\frac{1}{30}$	−15	+ 60	12·04	−1	+ 4·27
G	$\frac{1}{35}$	−15	+ 72·5	10·32	−1	+ 5·15
H	$\frac{1}{40}$	−15	+100	9·03	−1	+ 7·03
J	$\frac{1}{45}$	−15	+112·5	8·03	−1	+ 7·91
K	$\frac{1}{50}$	−15	+125	7·22	−1	+ 8·79
L	$\frac{1}{55}$	−15	+137·5	6·57	−1	+ 9·67
M	$\frac{1}{60}$	−15	+150	6·02	−1	+10·55
N	$\frac{1}{65}$	−15	+162·5	5·56	−1	+11·42
O	$\frac{1}{70}$	−15	+175	5·16	−1	+12·31
P	$\frac{1}{75}$	−15	+187·5	4·82	−1	+13·18
Q	$\frac{1}{80}$	−15	+200	4·52	−1	+14·05
R	$\frac{1}{85}$	−15	+212·5	4·25	−1	+14·94
S	$\frac{1}{90}$	−15	+225	4·01	−1	+15·82
T	$\frac{1}{95}$	−15	+237·5	3·80	−1	+16·70
V	$\frac{1}{100}$	−15	+250	3·61	−1	+17·60

N.B.—$\frac{1}{6}$ &c., on the Scale = 1 lb. pressure per sq. inch. 60·21, &c., m/m on the Scale = 1 Kilogramme per sq. c/m.



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53	.0000567		

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REFERENCE TO DIAGRAMS.

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